CALIFORNIA ENERGY
COMMISSION

Final Report Compilation for

Demand-Controlled Ventilation Assessment

TECHNICAL REPORT

October 2003
P-500-03-096-A8

Gray Davis, *Governor*

CALIFORNIA ENERGY COMMISSION

Prepared By:
Architectural Energy Corporation
Vernon A. Smith
Boulder, CO

Purdue University
James Braun
West Layfayette, IN

CEC Contract No. 400-99-011

Prepared For:
Christopher Scruton
Contract Manager

Nancy Jenkins
PIER Buildings Program Manager

Terry Surles
PIER Program Director

Robert L. Therkelsen
Executive Director

Acknowledgements

This project was a joint effort between Purdue University (Jim Braun, Kevin Mercer, and Tom Lawrence) and NIST (Andy Persily and Steven Emmerich). Todd Rossi and Doug Dietrich provided data collection services and field support, and David Jump with Nexant, Inc., and Lanny Ross with Newport Design Consultants provided field support as well. Honeywell Corporation was a match fund partner, providing DCV controllers and other hardware for the Project.

Preface

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The Program's final report and its attachments are intended to provide a complete record of the objectives, methods, findings and accomplishments of the Energy Efficient and Affordable Commercial and Residential Buildings Program. This attachment is a compilation of reports from Project 3.1, *Demand-Controlled Ventilation Assessment*, providing supplemental information to the final report (Commission publication #P500-03-096). The reports, and particularly the attachments, are highly applicable to architects, designers, contractors, building owners and operators, manufacturers, researchers, and the energy efficiency community.

This document is one of 17 technical attachments to the final report, consolidating eight research reports from Project 3.1:

- *Modeling and Testing Strategies for Evaluating Ventilation Load Reductions Technologies* (April 2001)
- *Description of Field Test Sites*. (Feb 2003, rev.)
- *State-of-the-Art Review of CO_2 Demand Controlled Ventilation Technology and Application*. NISTIR 6729 (Mar 2001)
- *VSAT – Ventilation Strategy Assessment Tool*. (Aug 2003)
- *Initial Cooling and Heating Season Field Evaluations for Demand-Controlled Ventilation*. (Feb 2003)
- *Simulations of Indoor Air Quality and Ventilation Impacts of Demand Controlled Ventilation in Commercial and Institutional Buildings*. NISTIR 7042 (Aug 2003)
- *Recommendations for Application of CO_2-Based Demand Controlled Ventilation: Proposed Design Requirements and Design Guidance for ASHRAE Standard 62 and Title 24*. (Aug 2003)
- *Evaluation of Demand Controlled Ventilation, Heat Pump Technology, and Enthalpy Exchangers*. (Aug 2003, rev)

The Buildings Program Area within the Public Interest Energy Research (PIER) Program produced this document as part of a multi-project programmatic contract (#400-99-011). The Buildings Program includes new and existing buildings in both the residential and the nonresidential sectors. The program seeks to decrease building energy use through research that will develop or improve energy-efficient technologies, strategies, tools, and building performance evaluation methods.

For the final report, other attachments or reports produced within this contract, or to obtain more information on the PIER Program, please visit www.energy.ca.gov/pier/buildings or contact the Commission's Publications Unit at 916-654-5200. The reports and attachments, as well as the individual research reports, are also available at www.archenergy.com.

Abstract

Project 3.1, *Demand-Controlled Ventilation Assessment*

A joint project between Purdue and NIST, investigated energy and cost savings associated with demand-controlled ventilation (DCV). In addition to energy and economic simulation and analysis supported by field experiments, the project provided a general study of indoor air quality implications of demand controlled ventilation.

- In most cases, the payback period associated with demand controlled ventilation with economizer override was less than two years.
- The greatest cost savings and lowest payback periods occur for buildings that have variable and unpredictable occupancy levels, such as auditoriums, gyms and retail stores.
- The greatest savings and lowest payback periods occur in the more extreme inland climates. Mild coastal climates have smaller savings and longer payback periods.

This document is a compilation of eight technical reports from the research.

DESCRIPTION OF FIELD TEST SITES
Revision for Walgreens Field Sites

Deliverables 2.1.1a, 2.1.1b, and 3.1.1a

Progress report submitted to:
Architectural Energy Corporation

For the Building Energy Efficiency Program
Sponsored by:
California Energy Commission

Submitted By:

Purdue University

Principal Investigator:	James Braun, Ph.D., P.E.
Research Assistants:	Tom Lawrence, P.E.
	Kevin Mercer
	Haorong Li

April 2001
Revision February 2003

Mechanical Engineering
1077 Ray W. Herrick Laboratories
West Lafayette, IN 47907-1077
(765) 496-6008
(765) 494-0787 (fax)

RAY W. HERRICK
LABORATORIES
PURDUE ENGINEERING

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

1. INTRODUCTION

Purdue University is under contract to Architectural Energy Corporation on behalf of the California Energy Commission (CEC) to conduct several research projects. This work is being done under the Building Energy Efficiency Program as part of the CEC's Public Interest Energy Research (PIER) Program.

1.1 Purdue Research Projects under this Program

The work at Purdue is focused on four specific projects and is being coordinated under the direction of Dr. James Braun, P.E. Each project covers different technologies or concepts that have shown promise for improving energy efficiency in building heating, ventilation and air conditioning (HVAC) systems. Specifically, the four projects that Purdue is working on include evaluations and studies of the following. (1) fault detection and diagnostics (FDD) of rooftop air conditioning units (Project 2.1); (2) demand controlled ventilation (DCV) assessment (Project 3.1); (3) assessment and field testing of ventilation recovery heat pumps (Project 4.2); and (4) night ventilation with building thermal mass (Project 3.2).

The first three of these projects are currently active, with the Project 3.2 scheduled to start in September of 2001. All four of the projects involve both theoretical analysis and field demonstration and evaluation. This report describes the field test sites selected for use in projects 2.1 and 3.1. Monitoring equipment has been installed at modular school room and restaurant field sites in Northern California. We have an agreement with the Walgreens Company to allow use of retail store sites in the Los Angeles metropolitan area, and installation is expected to being in August of 2001. An update to this report will be issued when the retail store installations are finalized.

1.2 Related Reports

This report describes the field test sites selected for use with the CEC PIER project. Other related reports submitted in parallel with this report are: (1) "Description of Laboratory Setup" and (2) "Modeling And Testing Strategies for Evaluating Ventilation Load Reduction Technologies.

The report “Description of Laboratory Setup” provides a description of the York rooftop unit and Honeywell Demand Controlled Ventilation system that are installed outside the Purdue Herrick Laboratory and the instrumentation used for monitoring the setup.. This setup follows closely the field site setups in California. The instrumentation includes measurement of system temperatures, pressures, relative humidities and carbon dioxide concentrations. The Laboratory Setup report covers in detail the setup and operation of the Virtual Mechanic hardware and ACRx ServiceTool Suite of monitoring software, both provided by Field Diagnostic Services. Finally, the report describes the general process for collecting and retrieving data downloaded from the field test sites.

The “Ventilation Strategy Analysis” report presents an overview of the modeling approach and input data to be used in evaluating the energy savings associated with several ventilation load reduction technologies. In addition, an overview of the preliminary test plan and field site monitoring setup for the heat pump heat recovery unit is given.

2. SELECTION OF FIELD TEST SITES

Projects 2.1 and 3.1 involve the use of 12 common field sites for evaluation of FDD and demand-controlled ventilation. In these two projects, field performance data will be obtained from heating/cooling units. Three different building types are being utilized in two different climate zones.

2.1 Criteria for selection of the building types

All of the Purdue projects are focused on small commercial buildings that utilize packaged air conditioning and heating equipment. The criteria used for selecting the types of buildings to include as field test sites focused on the typical building occupancy schedule, the building size and typical HVAC system installed, and the ability to identify multiple sites of similar design and construction within the same climate region. To reduce costs, the same test buildings are being used for the field studies in Projects 2.1 (fault detection and diagnostics) and 3.5 (demand-controlled ventilation). Earlier studies on demand-controlled ventilation indicated that the greatest benefits (in terms of energy savings) are possible with buildings that have variable occupancy schedules. Thus, the three building types selected for the field test sites are smaller retail stores, restaurants and schools. For each type of building, two nearly identical sites will be used in two different climates. This will allow comparative analysis of the energy savings associated with demand-controlled ventilation in terms of building type and climate. The fault detection and diagnostics project is focused strictly on small commercial packaged air conditioning units, so the field sites provide a range of equipment for demonstration and evaluation of this technology. A single site will be used to demonstrate a heat pump heat recovery unit. However, the data obtained from the demand-controlled ventilation sites can also be used to estimate savings for the heat pump heat recovery unit if it were installed in these additional sites.

A large number of modular schoolrooms are installed throughout the state of California. These rooms are all very similar in design and construction, and all typically use wall mounted heat pumps for heating and cooling. One advantage of the modular schoolroom for this study is that essentially identical rooms can be monitored side-by-side.

For the restaurant building type, the systems used to condition the children's play areas that are common in many fast food chains will be monitored. These rooms typically are self-contained, or nearly so, and only require one or two rooftop units for cooling and heating. By monitoring only the play areas in these restaurants, the study can gather data on spaces that have the greatest variability in occupancy, and also will eliminate the effects of the kitchen area and its associated ventilation systems.

The third building type selected is a small retail store. Small retail stores can have an extremely wide variation in occupancy patterns. Chain stores were considered for the study since essentially identical buildings can be found.

2.2 California Climate Types

Although California has a wide range of climate types, much of the state can be characterized as a Mediterranean climate. This climate type experiences warm, dry summers and temperate moist winters. The state also includes desert regions in southern California (such as Palm Springs) and coastal regions. The specific climate type for a given locality may vary significantly within a small distance due to the influence of factors such as topology and the proximity to the ocean. Some of the best examples of these variations occur in the San Francisco Bay area where the distance of just a few miles can lead to significant variations in rainfall patterns and sky conditions

2.3 Method for selecting sites

It is not possible within the scope of this project to evaluate the new technologies for all possible climate regions in California using field data. However, it will be possible to perform more extensive evaluations through simulation. For the field studies, representative buildings were selected in two different macroclimate types (coastal and inland). In addition, some of the selected sites are in northern California and some are in southern California, which gives as wide a range of location and climate type as practical within the context of these projects. The inland sites vary from the Mediterranean climate type of the Central Valley around Sacramento to the desert regions around Palm Springs. Although it was not possible to have field sites for all technologies in all climate regions, the areas selected for study represent those with the greatest concentration of population and commercial development.

Before the projects officially started, contacts were made with the owners of potential building sites within the school, restaurant and retail store categories. The identification of sites has been a time consuming process that has required the help of several of the participating organizations, including Honeywell, Schiller Associates, Carrier Corporation, Southern California Edison, and Architectural Energy Corporation.

The first buildings identified were schools. During the summer of 2000, contacts were made and meetings held with representatives of the Oakland Unified School District and the Woodland Joint Unified School District. Woodland is approximately 20 miles west of Sacramento and represents an inland climate type. The monitoring systems were installed at two rooms located side-by-side at each of the two school districts in December of 2000. More details on these sites are contained later in this report.

The restaurant building type is represented by two franchisee owned McDonald's stores in the Sacramento area and by two corporate owned stores on the southeastern San Francisco Bay area. These stores have PlayPlace areas with similar construction and HVAC system installations, although it was not possible to find stores with identical design and sun orientation. Sun orientation can be particularly important for the PlayPlace areas, since they typically include a large percentage of glass area. Monitoring equipment was installed in the Sacramento McDonalds during the middle of March, 2001. In the San Francisco Bay Area, a representative of McDonalds corporate office identified two stores for inclusion in our study that will be the best fit for our needs. Monitoring equipment were installed in May of 2001 at these two stores. More details on these sites are also given in the later sections of this report.

The retail stores are in Southern California. The Walgreens corporation has agreed to our using their stores as part of this program. Monitoring systems are installed at stores located in Rialto (near Riverside) and Anaheim. The Rialto store is located in a near desert climate, while Anaheim is a more coastal climate type.

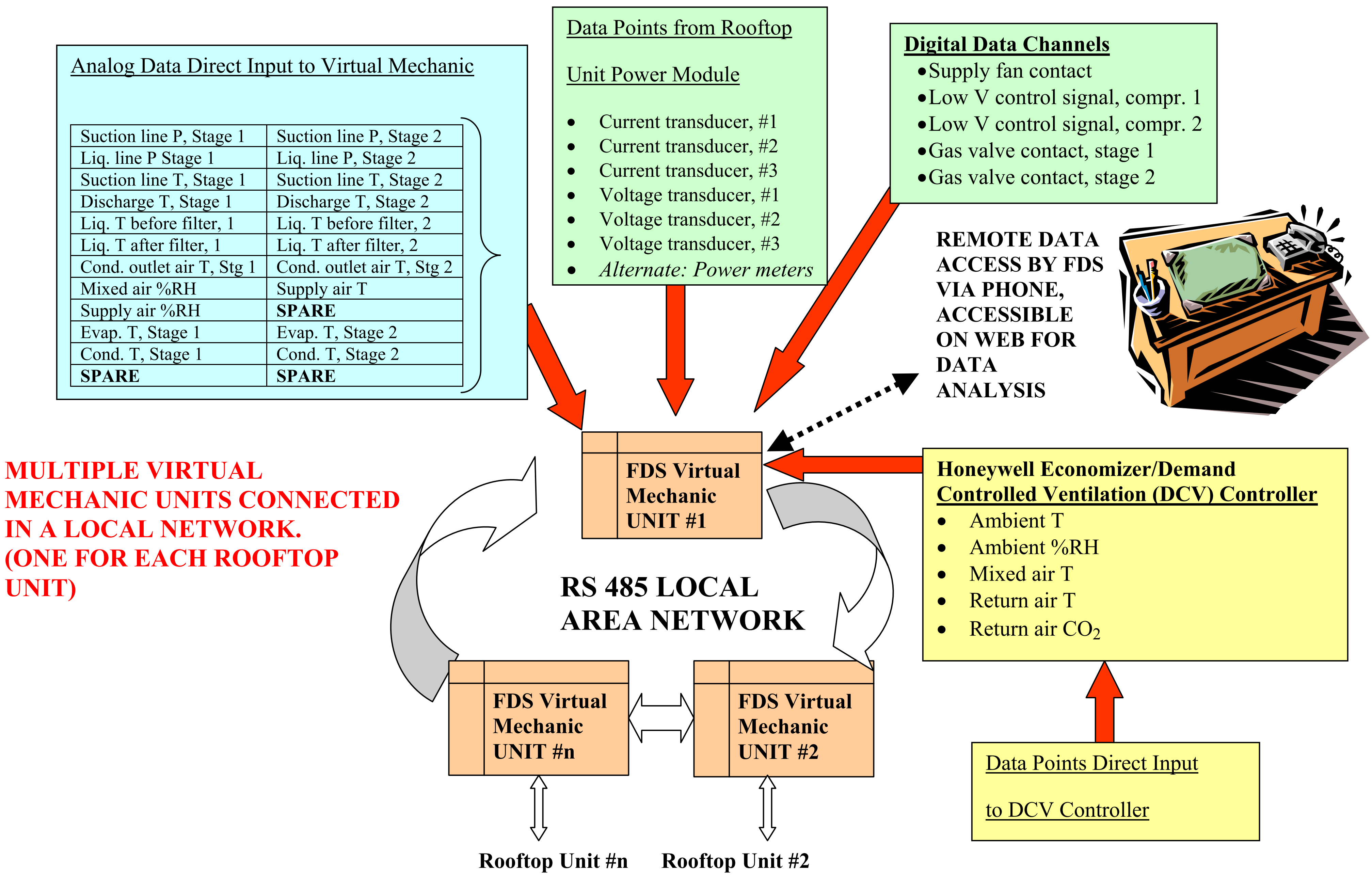
3. DESCRIPTION OF FIELD TEST SITES

Figure 1 presents a general overview of how data are monitored and collected from the field sites. Proprietary equipment from Honeywell controls ventilation dampers using economizer and demand-control ventilation algorithms. The Honeywell controller incorporates sensors to measure ambient temperature and humidity, return air temperature and carbon-dioxide concentration, and mixed air temperature. Additional sensors are installed to monitor other air state variables, refrigerant states, power consumption, and operational status. The primary data acquisition is accomplished using hardware from Field Diagnostics Services (FDS) called the Virtual Mechanic (VM). The VM communicates with the Honeywell controller across an RS485 network to obtain sensor information and to change control strategies. The additional sensors are wired directly to the VM. Data are sampled at approximately 5-minute intervals and are stored in the VM. For some field sites, multiple VMs are employed for multiple packaged air conditioners. Data are downloaded each day using cell phones connected to the master Virtual Mechanic at each test site.

A detailed description of the field test sites is provided in the following subsections. Some of the detailed technical information needed to simulate the performance of the different technologies for these buildings will be compiled later in the project. This section contains information on the following test sites:

- Modular School Rooms – Inland Climate Type
- Modular School Rooms – Coastal Climate Type
- Fast Food Restaurants – Inland Climate Type
- Fast Food Restaurants – Coastal Climate Type
- Retail Stores – Inland Climate
- Retail Stores – Coastal Climate

Figure 1 – Field Test Sites Data Collection and Communication Overview



BUILDING TYPE: **Modular School Rooms**

Inland Climate Locations

ADDRESS: Gibson Elementary School
312 Gibson Road
Woodland, CA 95695
(530) 662-3944

EQUIPMENT INSTALLATION DATE: December 14-19, 2000

CELL PHONE NUMBER: (765) 427-0311

DETAILED BUILDING DESCRIPTION:

Floor Area	20 feet by 40 feet (800 sq. ft.)
Building Orientation	East – West
Wall Construction	Walls are 2x4 stud construction with R-11 insulation. Internal walls have ¾" vinyl covered fiberboard over $^5/_8$" gypsum wallboard.
Windows/ Shading	Wood panel exterior with no windows on south or north sides. East and west sides have one 4' x 8' window, with door on east side. Two-foot overhang on west wall and three-foot overhang on east wall entrance area. Windows are double-pane with ¼" air gap.
Roof/Ceiling Construction	Flat roof with reflective paint coating. Roof has R-19 insulation. Interior drop ceiling is 8' above occupied space with t-bar 18" below the roof.
Floor	Crawl space below is ventilated with R-11 insulation below floor.
Lighting	10 sets of fluorescent lights, 120 W each with magnetic ballast.
Other Loads and Equipment	One desktop computer and one small refrigerator.
Occupancy Patterns	8:30 am to 3:00 pm weekdays. Usually one or two hours on Saturday mornings. The rooms are occupied by 15-20 small children per room, plus teacher. (These are kindergarten – first grade rooms.)

<u>*Gibson School (Cont'd)*</u>

Woodland School Site –
Rear View Looking East

Woodland School Site –
Front View Looking West

Each building (modular school room) has its own packaged air conditioner/heat pump. Two side-by-side units have been retrofit with the Honeywell economizer and demand control ventilation system and fully instrumented. Two VMs are networked together with one of units linked to a cell phone. The heat pump units were originally set up for fixed percentage of outdoor air, and did not have outdoor air flow control dampers. It was estimated that, based on the installation configuration, the airflow control was set up for approximately 15% outdoor air at these sites before the retrofit.

HEATING / AIR CONDITIONING EQUIPMENT:

Each building (room) has a sidewall-mounted heat pump as described in the table below.

Manufacturer	Bard Manufacturing
Model	WH 421-A
Nominal Cooling Capacity	3½ Tons
Number of Stages	1
SEER / HSPF	10.0 / 6.8
Supplemental Heating Capacity	10 kW nominal electric resistance heater.
Electrical	Single phase, 220 V
Supply Fan Rating	1400 cfm @ 0.3”

TEST INSTRUMENTATION:

Table 1 lists the input data channels used at the modular schoolrooms. The same data list is used at both the Woodland and Oakland school sites.

Table 1 – Data List for Modular School Room Field Test Sites

Channel #	Data Point
Power Transducer Channels	
1	Unit voltage
2	Compressor 1 voltage
3	Common
4	Unit total current
5	Compressor 1 current
6 - 8	Spare - Not Used
Other Analog Input Data	
9	Suction line pressure, Stage 1
10	Liquid line pressure, Stage 1
11 - 14	Spare - Not Used
15	Mixed air temperature
16	Return air temperature
17	Supply air temperature, before heater
18	Supply air temperature, after heater
19	Condenser inlet air temperature
20	Condenser outlet air temperature
21	Suction line temperature, Stage 1
22	Discharge line temperature, Stage 1
23	SPARE - Used as additional ambient T
24	SPARE - Used as additional ambient T
25	Evaporation temperature, Stage 1
26	Condensation temperature, Stage 1
27- 32	Spare - Not Used
Calculated Data Channels	
33-50	NOT USED
51	Honeywell DCV indoor (and outdoor) CO2 conc.
52	Honeywell DCV mixed air temperature
53	Honeywell DCV return air temperature
54	Honeywell DCV return / outdoor humidity
55	Honeywell DCV outdoor air temp & damper position
56	Honeywell DCV minimum damper position
57	superheat, stage 1
58	subcooling, stage 1
59	evaporating temperature, stage 1
60	condensing temperature, stage 1
61	condensing temperature over ambient (CT-AIC), stage 1
62	superheat, stage 2
63	subcooling, stage 2
64	evaporating temperature, stage 2
65	condensing temperature, stage 2
66	condensing temperature over ambient (CT-AIC), stage 2

Table 1 – Data List for Inland Modular School Room Field Test Site (Cont'd)

Channel	Data Point
67	evaporator temperature difference (RA-SA)
68	NOT USED
69	NOT USED
70	unit power (kW)
71	unit KWh
72	unit MWh
73	compressor 1 power (kW)
74	compressor 1 KWh
75	compressor 1 MWh
76	compressor 2 power (kW)
77	compressor 2 KWh
78	compressor 2 MWh
79	digital input 1, supply fan, run time (8 hours)
80	digital input 1, supply fan, run time (seconds)
81	digital input 2, cooling 1, run time (8 hours)
82	digital input 2, cooling 1, run time (seconds)
83	digital input 3, cooling 2, run time (8 hours)
84	digital input 3, cooling 2, run time (seconds)
85	digital input 4, heat 1, run time (8 hours)
86	digital input 4, heat 1, run time (seconds)
87	digital input 5, heat 2, run time (8 hours)
88	digital input 5, heat 2, run time (seconds)
89	digital input 6 run time (8 hours)
90	digital input 6 run time (seconds)
91	time since reset accumulators (8 hours)
92	time since reset accumulators (seconds)
93	up time (8 hours)
94	up time (seconds)
95	board temperature (F)
96	board battery voltage (V)
Digital Channels	
1	Supply fan contact (fan on / fan off)
2	Low voltage control signal for compressor contact
3	Spare
4	Heat on
5	Electric heat
6	

BUILDING TYPE: **Modular School Rooms**

Coastal Climate Location

ADDRESS: Fremont High School
4610 Foothill Blvd.
Oakland, CA
(510) 879-3020

EQUIPMENT INSTALLATION DATE: December 19-21, 2000

CELL PHONE NUMBER: (765) 427-0325

DEtAILED DESCRIPTION:

Floor Area	20 feet by 40 feet (800 sq. ft.)
Building Orientation	East – West
Wall Construction	Walls are 2x4 stud construction with R-11 insulation. Internal walls have ¾" vinyl covered fiberboard over $^5/_8$" gypsum wallboard.
Windows/ Shading	Wood panel exterior with no windows on south or north sides. East and west sides have one 4' x 8' window, with door on east side. Two-foot overhang on west wall and three-foot overhang on east wall entrance area. Windows are double-pane with ¼" air gap.
Roof/Ceiling Construction	Flat roof with reflective paint coating. Roof has R-19 insulation. Interior drop ceiling is 8' above occupied space with t-bar 18" below the roof.
Floor	Crawl space below is ventilated with R-11 insulation below floor.
Lighting	Approximately 10 sets of fluorescent lights, 120 W each with magnetic ballast.
Other Loads and Equipment	One desktop computer. (To be verified)
Occupancy Patterns	8:30 am to 3:00 pm weekdays. The rooms are occupied by 15-20 high school students per classroom.

Fremont High School (Cont'd)

Oakland School Site (Fremont High School) - View Looking Along North Walls

Each building (modular school room) has its own packaged air conditioner/heat pump. Two side-by-side units have been retrofit with the Honeywell economizer and demand control ventilation system and fully instrumented. Two VMs are networked together with one of units linked to a cell phone. The heat pump units were originally set up for fixed percentage of outdoor air, and did not have outdoor air flow control dampers. It was estimated that, based on the installation configuration, the airflow control was set up for approximately 15% outdoor air at these sites before the retrofit.

HEATING / AIR CONDITIONING EQUIPMENT:

Each building (room) has a sidewall-mounted heat pump manufactured by Bard Industries, Model WH 421A. These are the same units as used at the Woodland school site. The units are contained within a fenced off area on the north end of the buildings.

Nominal Cooling Capacity	3½ Tons
SEER / HSPF	10.0 / 6.8
Heating Capacity	10 kW nominal electric resistance heater. Note: The electrical resistance heaters are not functioning for these rooms.
Electrical	Single phase, 220 V
Supply Fan Performance	1400 cfm @ 0.3”

TEST INSTRUMENTATION:

The Fremont school site uses the same data point list given in Table 1 for the Woodland schools.

BUILDING TYPE: **Fast Food Restaurants**

Inland Climate Locations

ADDRESS:

McDonalds Restaurant
2434 Watt Ave.
Sacramento, CA 95821
(916) 971-0244

3560 Bradshaw Road
Sacramento, CA 95827
(916) 361-8186

CONTACT:

Mike Godlove (Owner)
2508 Garfield Ave
Carmichael, CA 95608
(916) 483-6065

EQUIPMENT INSTALLATION DATE: March 12-14, 2001

CELL PHONE NUMBERS: (765) 427-7714 and 427-7919

DEtAILED DESCRIPTION:

Equipment at two nearly identical McDonald's PlayPlaces in Sacramento have been retrofit with the Honeywell economizer and demand control ventilation system and fully instrumented. Each system has its own dedicated VM with a cell phone for data transmission. The Watt Avenue site has a slightly smaller floor area (approximately 20 square feet less take from two corners). The following subsections give some details on the building construction and operation. Additional details will be obtained later.

Sacramento Area McDonalds PlayPlace Construction (Watt Avenue and Bradshaw Road)

Floor Area	Approximately 20 feet by 30 feet (600 sq. ft.) that is for the most part isolated from the dining and cooking areas.
Building Orientation	Primary axis for this room is North - South. Major glass surfaces on the East and South walls. West face is interior wall shared with the dining area.
Wall Construction	"Stucco" exterior covering.
Windows/ Shading	Major glass surfaces on the East and South walls. West face is interior wall shared with the dining area. Some window area on North wall. No exterior shading. Windows are tinted with double pane, ¼" air gap construction.
Roof/Ceiling Construction	Flat roof with light colored asphalt coating.
Floor	Tile on slab construction.
Lighting	Approximately six sets of fluorescent lights, with four bulbs each with magnetic ballast.
Other Loads and Equipment	Some air exchange with dining area and outdoor air via door in the common vestibule. Ceiling fans keep air in motion.
Occupancy Patterns	PlayPlace hours are: 9 am to 9:30 pm. Occupancy varies from 0 to a maximum of approximately 40.

Watt Avenue (Sacramento Area) McDonalds PlayPlace Pictures

Interior view of Watt Avenue McDonalds PlayPlace Area showing location of return air and supply air ducts.

**Watt Avenue McDonalds –
View Looking Southwest**

**Watt Avenue McDonalds –
Rooftop Units Undergoing Equipment Installation**

Bradshaw Road (Sacramento Area) McDonalds PlayPlace Pictures

Interior view of Bradshaw Road McDonalds PlayPlace Area

Bradshaw Road McDonalds – View Looking Northwest

Bradshaw Road McDonalds – Rooftop Units Undergoing Equipment Installation

HEATING / AIR CONDITIONING EQUIPMENT:

Each PlayPlace uses rooftop-mounted units for providing heating, cooling and ventilation air to the room. The two sites differ in the number of rooftop units used, with the Watt Avenue building using one two-stage unit and the Bradshaw Road building using two smaller single-stage units. According to York International's regional support representative, the units are custom designed for supply to McDonalds Corporation for the PlayPlace areas. The following tables describe the units used at each site. Since they are custom designs, published performance ratings and other technical details were not readily available. This information will be obtained later.

Watt Avenue

Manufacturer	York International
Model	D3CG120N20025MKD
Nominal Cooling Capacity	10 Tons
Number of Stages	2
SEER / HSPF	TBD
Heating Capacity	200,000 Btu/hr nominal output
Electrical	Three phase, 220 V
Supply Fan Performance	4,000 cfm manufacture rated

Bradshaw Road

Manufacturer	York International
Model	D1CG072N07925ECC
Nominal Cooling Capacity	6 Tons
Number of Stages	1
SEER / HSPF	TBD
Heating Capacity	100,000 Btu/hr nominal output
Electrical	Three phase, 220 V
Supply Fan Performance	2,400 cfm manufacture rated (each)

TEST INSTRUMENTATION:

Tables 2 and 3 list the data channels used at the restaurants. A slightly different list is required for each site since the HVAC equipment setup is different. In particular, the Watt Avenue site has one larger (10 ton) unit with 2-stage cooling to condition the entire room. The Bradshaw Road site, on the other hand, has two smaller (6 ton) single-stage cooling units operating in parallel. Instrumentation for fault detection and diagnostics and monitoring was set-up for one rooftop unit per site, as originally planned in the project proposal stage. Therefore, one unit at the Bradshaw Road site was fully instrumented for both FDD and DCV purposes, while the second unit was instrumented only for the purposes of collecting data for the DCV project. The Watt Avenue site has only one rooftop unit and was fully instrumented according to the standard data list. All data will be collected using one Virtual Mechanic at each site.

Table 2 – Data List for Inland Restaurant Field Test Site (Watt Avenue)

Channel #	Data Point
SENSOR CHANNELS	
Power Transducer Channels	
1	Unit voltage
2	Compressor 1 voltage
3	Compressor 2 voltage
4	Unit total current
5	Compressor 1 current
6	Compressor 2 current
Other Analog Input Data	
7	SPARE - Not used
8	SPARE - Not used
9	Suction line pressure, Stage 1
10	Discharge pressure, Stage 1
11	Suction line pressure, Stage 2
12	Discharge pressure, Stage 2
13	SPARE - Not used
14	SPARE - Not used
15	Mixed air temperature
16	Return air temperature
17	Supply air temperature, before heater
18	Supply air temperature, after heater
19	Condenser inlet air temperature
20	Condenser outlet air temperature
21	Suction line temperature, Stage 1
22	Discharge line temperature, Stage 1
23	Liquid line temperature before filter/drier, Stage 1
24	Liquid line temperature after filter/drier, Stage 1
25	Evaporation temperature, Stage 1
26	Condensation temperature, Stage 1
27	Suction line temperature, Stage 2
28	Discharge line temperature, Stage 2
29	Liquid line temperature before filter/drier, Stage 2
30	Liquid line temperature after filter/drier, Stage 2
31	Evaporation temperature, Stage 2
32	Condensation temperature, Stage 2
Calculated Data Channels	
33-50	NOT USED
51	Honeywell DCV indoor (and outdoor) CO2 conc.
52	Honeywell DCV mixed air temperature
53	Honeywell DCV return air temperature
54	Honeywell DCV return / outdoor humidity
55	Honeywell DCV outdoor air temp & damper position

Table 2 – Data List for Inland Restaurant Field Test Site (Watt Avenue) – Cont'd

Channel	Data Point
56	Honeywell DCV minimum damper position
57	superheat, stage 1
58	subcooling, stage 1
59	evaporating temperature, stage 1
60	condensing temperature, stage 1
61	condensing temperature over ambient (CT-AIC), stage 1
62	superheat, stage 2
63	subcooling, stage 2
64	evaporating temperature, stage 2
65	condensing temperature, stage 2
66	condensing temperature over ambient (CT-AIC), stage 2
67	evaporator temperature difference (RA-SA)
68	NOT USED
69	NOT USED
70	unit power (kW)
71	unit KWh
72	unit MWh
73	compressor 1 power (kW)
74	compressor 1 KWh
75	compressor 1 MWh
76	compressor 2 power (kW)
77	compressor 2 KWh
78	compressor 2 MWh
79	digital input 1, supply fan, run time (8 hours)
80	digital input 1, supply fan, run time (seconds)
81	digital input 2, cooling 1, run time (8 hours)
82	digital input 2, cooling 1, run time (seconds)
83	digital input 3, cooling 2, run time (8 hours)
84	digital input 3, cooling 2, run time (seconds)
85	digital input 4, heat 1, run time (8 hours)
86	digital input 4, heat 1, run time (seconds)
87	digital input 5, heat 2, run time (8 hours)
88	digital input 5, heat 2, run time (seconds)
89	digital input 6 run time (8 hours)
90	digital input 6 run time (seconds)
91	time since reset accumulators (8 hours)
92	time since reset accumulators (seconds)
93	up time (8 hours)
94	up time (seconds)
95	board temperature (F)
96	board battery voltage (V)

Table 2 – Data List for Inland Restaurant Field Test Site (Watt Avenue) – Cont'd

Digital Channels

1	Supply fan contact (fan om / fan off)
2	Low voltage control signal for compressor 1 contact
3	Low voltage control signal for compressor 2 contact
4	Heating 1
5	Heating 2
6	

Table 3 – Data List for Inland Restaurant Field Test Site (Bradshaw Road)

Channel #	Data Point
SENSOR CHANNELS	
Power Transducer Channels	
1	Unit 1 input voltage
2	Compressor voltage, Unit 1
3	Unit 2 input voltage
4	Unit 1 total current
5	Compressor current, Unit 1
6	Unit 2 total current
Other Analog Input Data	
7	**SPARE - Not used**
8	**SPARE - Not used**
9	Suction line pressure, Unit 1
10	Discharge pressure, Unit 1
11	**SPARE - Not used**
12	**SPARE - Not used**
13	**SPARE - Not used**
14	**SPARE - Not used**
15	Mixed air temperature - Unit 1
16	Return air temperature - Unit 1
17	Supply air temperature, before heater - Unit 1
18	Supply air temperature, after heater - Unit 1
19	Condenser inlet air temperature - Unit 1
20	Condenser outlet air temperature - Unit 1
21	Suction line temperature - Unit 1
22	Discharge line temperature - Unit 1
23	Liquid line temperature before filter/drier - Unit 1
24	Liquid line temperature after filter/drier - Unit 1
25	Evaporation temperature - Unit 1
26	Condensation temperature - Unit 1
27	**SPARE - Not used**
28	**SPARE - Not used**
29	Mixed air temperature - Unit 2
30	Mixed air humidity - Unit 2
31	Supply air temperature - Unit 2
32	Supply air humidity - Unit 2
CALCULATED DATA CHANNELS	
33-50	NOT USED
51	Honeywell DCV indoor (and outdoor) CO2 conc.
52	Honeywell DCV mixed air temperature
53	Honeywell DCV return air temperature
54	Honeywell DCV return / outdoor humidity
55	Honeywell DCV outdoor air temp & damper position

Table 3 – Data List for Inland Restaurant Field Test Site (Bradshaw Road) – Cont'd

Channel	Data Point
56	Honeywell DCV minimum damper position
57	superheat, stage 1
58	subcooling, stage 1
59	evaporating temperature, stage 1
60	condensing temperature, stage 1
61	condensing temperature over ambient (CT-AIC), stage 1
62	NOT USED
63	NOT USED
64	NOT USED
65	NOT USED
66	NOT USED
67	NOT USED
68	NOT USED
69	NOT USED
70	unit power (kW)
71	unit KWh
72	unit MWh
73	compressor 1 power (kW)
74	compressor 1 KWh
75	compressor 1 MWh
76	compressor 2 power (kW)
77	compressor 2 KWh
78	compressor 2 MWh
79	digital input 1, supply fan, run time (8 hours)
80	digital input 1, supply fan, run time (seconds)
81	digital input 2, cooling 1, run time (8 hours)
82	digital input 2, cooling 1, run time (seconds)
83	digital input 3, cooling 2, run time (8 hours)
84	digital input 3, cooling 2, run time (seconds)
85	digital input 4, heat 1, run time (8 hours)
86	digital input 4, heat 1, run time (seconds)
87	digital input 5, heat 2, run time (8 hours)
88	digital input 5, heat 2, run time (seconds)
89	digital input 6 run time (8 hours)
90	digital input 6 run time (seconds)
91	time since reset accumulators (8 hours)
92	time since reset accumulators (seconds)
93	up time (8 hours)
94	up time (seconds)
95	board temperature (F)
96	board battery voltage (V)

Table 3 – Data List for Inland Restaurant Field Test Site (Bradshaw Road) – Cont'd

Digital Channels	
1	Supply fan contact (fan om / fan off)
2	Low voltage control signal for compressor contact
3	Spare
4	Heating
5	Spare
6	

BUILDING TYPE: **Fast Food Restaurants**

Coastal Climate Locations

ADDRESS:

99 N. Milpitas Blvd.
Milpitas, CA 95035
(408) 263-0181

1620 Storbridge Ave.
Castro Valley, CA 94546
(510) 537-9566

CONTACT:

Paul Martin
(408) 422-2339

EQUIPMENT INSTALLATION DATE: May 2001

CELL PHONE NUMBERS:

(765) 427-2988
(765) 427-3052

DETAILED DESCRIPTION:

The PlayPlace areas at these two sites are not as close in design and orientation as are the two Sacramento sites. This is a compromise in order to get two sites that are reasonably close together and in a similar coastal climate zone. Both restaurants are located south of Oakland on the east edge of the San Francisco Bay and have a floor space of around 1300 square feet, which is larger than the PlayPlace areas at the two Sacramento stores. The Castro Valley restaurant is oriented with its main glass area facing west. The Milpitas store, however, contains a larger glass area and is oriented facing north. The following subsections contain some descriptions of the room construction and heating/cooling equipment for these two coast climate restaurant sites. Additional details of the construction and building operation will be obtained later.

Castro Valley (San Francisco Bay Area) McDonalds PlayPlace Construction

Floor Area	Approximately 26 feet by 50 feet (1300 sq. ft.) that is isolated from the dining and cooking areas by an interior glass wall with two doors.
Building Orientation	Primary axis for this room is northwest - southeast. The long axis glass surface area faces southwest, with the smaller sides facing northwest and southeast. Northeast wall is interior wall shared with the dining area.
Wall Construction	"Stucco" exterior covering.
Windows/ Shading	Windows are tinted with double pane, ¼" air gap construction. Overhang of 24" at top that provides minimal shading. Total glass area of about 490 sq. ft. on southwest wall and 195 sq. ft. each on the northwest and southeast walls.
Roof/Ceiling Construction	Flat roof with light colored asphalt coating.
Floor	Tile on slab construction.
Lighting	Total of 26 fixtures of 48" fluorescent lights, with four bulbs each with magnetic ballast. Several had missing bulbs, only approximately 80% of bulbs in place.
Other Loads and Equipment	One TV and four video games. Ceiling fans keep air in motion.
Occupancy Patterns	PlayPlace operating hours are 9am – 9pm. During visit on a Sunday afternoon, occupied by approximately 70 children and adults.

Milpitas (San Francisco Bay Area) McDonalds PlayPlace Construction

Floor Area	Approximately 24 feet by 50 feet with 6' by 6' corner that shares internal wall with kitchen storage. Total floor is approximately 1170 sq. ft. Zone is isolated from the dining and cooking areas by an interior glass wall with two doors.
Building Orientation	Primary axis for this room is east - west. The long axis glass surface area faces north, with the smaller sides facing west and east. South wall is interior wall shared with the dining area.
Wall Construction	"Stucco" exterior covering.
Windows/ Shading	Windows are tinted with double pane, ¼" air gap construction. Overhang of 24" at top that provides minimal shading. Exterior walls are essentially floor to ceiling covered in glass. Total glass area of about 1000 sq. ft. on north wall, 480 sq. ft. on the east wall and 360 sq. ft. on the west wall.
Roof/Ceiling Construction	Flat roof with light colored asphalt coating.
Floor	Tile on slab construction.
Lighting	Total of 19 fixtures of 48" fluorescent lights, with four bulbs each with magnetic ballast.
Other Loads and Equipment	No TVs or video games. Ceiling fans keep air in motion.
Occupancy Patterns	PlayPlace operating hours are 8am – 9pm.

Castro Valley McDonalds PlayPlace Pictures

Interior view of Castro Valley McDonalds PlayPlace Area.

Castro Valley McDonalds – View Looking Southeast

Castro Valley McDonalds PlayPlace Area. York Rooftop Unit

Milpitas McDonalds PlayPlace Pictures

Interior view of Milpitas McDonalds PlayPlace Area (NW Corner)

Milpitas McDonalds – View Looking Southeast

Milpitas McDonalds PlayPlace Area. Two York Rooftop Units

HEATING / AIR CONDITIONING EQUIPMENT:

Each building (room) uses rooftop-mounted units for providing heating, cooling and ventilation air to the room. The two sites differ in the number of rooftop units used. Just like the two restaurants in Sacramento, one restaurant uses one two-stage York rooftop unit (Castro Valley) and the other (Milpitas) uses two smaller single-stage units. The units are of the same series that were designed and built specifically for the McDonalds PlayPlace areas. The following tables describe the units used at each site. Since they are more or less custom design, published performance ratings and other technical details were not readily available.

Castro Valley

Manufacturer	York International
Model	D4CG150N16525MDB
Nominal Cooling Capacity	12 Tons
Number of Stages	2
SEER / HSPF	TBD
Heating Capacity	204,000 Btu/hr nominal output
Electrical	Three phase, 220 V
Supply Fan Performance	4,000 cfm manufacture rated

Milpitas

Manufacturer	York International
Model	D1CG072N09925C
Nominal Cooling Capacity	6 Tons
Number of Stages	1
SEER / HSPF	TBD
Heating Capacity	125,000 Btu/hr nominal output
Electrical	Three phase, 220 V
Supply Fan Performance	2,400 cfm manufacture rated (each)

TEST INSTRUMENTATION:

Similar test instrumentation will be used as for the Sacramento McDonalds. The system at the restaurant with only one rooftop unit (Castro Valley) will be fully instrumented for both FDD and DCV studies, like the Watt Avenue site in Sacramento. The data list is presented in Table 2. The Milpitas site is analogous to the Bradshaw Road store in Sacramento, whereby one unit will be fully instrumented for both FDD and DCV purposes, while the second unit will be instrumented only for the purposes of collecting data for the DCV project. Table 3 provides this data list. All data will be collected using one VM at each site.

BUILDING TYPE: **Retail Store**

ADDRESS:

Inland Climate Location	Walgreens 550 S. Riverside Rialto, CA
Contact:	Gabriel Reyes (Store Manager) (709) 874-6600
Coastal Climate Location	Walgreens 946 S. Brookhurst Anaheim, CA
Contact:	Lee Anderson (Store Manager) (714) 520-5444

EQUIPMENT INSTALLATION DATES:

Rialto Store:	VM Monitoring Equipment: August 1-5, 2001 Functioning Honeywell Controls: June, 2002
Anaheim Store:	VM Monitoring Equipment: June, 2002 Functioning Honeywell Controls: Fall 2002

CELL PHONE NUMBERS: Dedicated land phone lines were installed in August 2002 to replace the cell phone arrangement.

DETAILED BUILDING DESCRIPTION: Rialto Store (Common Design)

Floor Area	100 feet by 90 feet (9,000 sq. ft.) in retail store space, 40 feet by 20 feet in the pharmacy. An additional 35 feet by 90 feet of backroom storage and 20 feet by 100 feet for office and equipment that is not part of the DCV study.
Building Orientation	Generally north - south, with front door on northeast corner.
Wall Construction	Brick and stucco exterior.
Windows/ Shading	A total of 20 windows on the two exterior walls to the retail store area. Windows are 5 feet by 8 feet, tinted, double-pane with ¼” air gap. Windows are on the east and north walls. A five-foot overhang covers the sidewalk and shades the exterior windows.

Roof/Ceiling Construction	Flat roof with light store coating.
Floor	Floor tiles over concrete slab.
Lighting	Retail store has total of 170 fixtures with 2 bulbs, 8-foot long fluorescent lights. Pharmacy has 33 fixtures of 2 bulb, four-foot long fixtures.
Other Loads and Equipment	Refrigerated drink and food open to store, 25 feet linear feet. Freezer section with doors, 20 feet long. Photo processing machine plus two cash registers.
Occupancy Patterns	Store hours are 8 am to 10 pm, seven days a week.

<u>HEATING / AIR CONDITIONING EQUIPMENT:</u>

Four rooftop heat pumps condition the retail store space and one additional unit is dedicated to the pharmacy area. A separate unit is installed at the store to condition the storage room, but since this is an isolated area not normally occupied, it is not part of the DCV installation study. The rooftop units are manufactured by Trane.

Manufacturer	Trane
Model	WFD090C30BBC - Retail Store WFD075C30BBC - Pharmacy
Nominal Cooling Capacity	Retail store units - 7½ tons Retail store units - 6¼ tons
Number of Stages	1
SEER / HSPF	8.9 EER
Electrical	Three phase, 208 V
Supply Fan Performance	2,500 nominal supply airflow @ 0.5 in. w.c. - 6¼ tons 3,000 nominal supply airflow @ 0.5 in. w.c. - 7½ tons

TEST INSTRUMENTATION:

Similar test instrumentation is used as for the McDonalds sites. Individual VM monitoring systems are installed for each rooftop unit, and networked together to one master VM that communicates via the cell phone. These rooftop units are single stage compressor systems, and the same monitoring data as listed in Table 3 are used.

Trane rooftop heat pump installed on Walgreens Rialto store

4. TESTING PLAN

This test plan as outlined below was set up during the initial phases of the project. The test plan has changed as the result of equipment installation schedules and problems. The field sites were rotated more regularly between demand control ventilation ON and OFF remotely using procedures developed by Field Diagnostic Services.

Data is downloaded on a daily basis using cell phones connected to the master Virtual Mechanic at each test site. The data monitoring and collection process was outlined earlier in this report in Figure 1.

There are separate test plans for the two projects that share the 12 field test site buildings.

Project 2.1: Fault Detection and Diagnostics

A testing plan for this project is included in a separate report being submitted by Purdue for deliverable 2.1.1b. This report is titled, "Description of Laboratory Setup" and was described in Section 1.2 above.

Project 3.1: Demand Controlled Ventilation

The following is a general overview of the testing plan for Project 3.1. The separate report titled "*Modeling and Testing Strategies for Evaluating Ventilation Load Recovery Technologies*" being submitted by Purdue describes how the data being collected will be analyzed.

Key parameters to measure for this project are:

- Unit power consumption for the compressors and fans.
- Energy input during heating mode. This will be expressed either in terms of compressor and electrical resistance heater power for the sites with heat pump heating, or in terms of natural gas usage for rooftop units with heating.
- Total cycle time for compressor (and heater) operation.
- Levels of carbon dioxide in the occupied space.

- Temperature and humidity levels for the ambient air, mixed air, supply air and the conditioned space.

The following is a general outline of the data gathering and test plan.

SCHOOLS:

March – May 2001: Monitor building performance for each of the four schoolrooms. Use this data to build baseline data for each room.

May – June, 2001: For the remaining part of this school year, set up one building at each site to run in Demand Controlled Ventilation (DCV) mode and the other building with the standard economizer mode. During this time visit each room and characterize the nominal usage patterns, etc.

Summer, 2001 (June-August): If the rooms are not to be occupied regularly during the summer months when regular school is not in session (mid-June to early September), then set up each room to operate in one common mode. Since the units at both school sites were setup for fixed outdoor air ventilation rates originally, we will duplicate that situation with the same percentage of outdoor air for each room. This will allow for a full characterization of the building thermal performance and any baseline differences between rooms at each site.

Fall, 2001: Around the beginning of the new school year, the control strategy will be changed to include one building on DCV and the other on a fixed ventilation rate. The fixed ventilation rate will be for the maximum setting required for schoolroom occupancy as determined by ASHRAE Standard 62. The control strategies will be reversed from that during the initial cooling season monitoring time (May to June).

November 2001 – January 2002: Maintain the same control strategy for each building for the beginning of the heating season.

January 2002 – March 2002: Reverse ventilation control strategies between the buildings at each climate type. Do this during a site visit in late December 2001 or early January

2002, or remotely if possible. Change back to the same settings for each room as with the first cooling season phase of May-June, 2001.

<u>RESTAURANTS:</u>

March – May 2001: Monitor building performance for each of the restaurants using one common ventilation control strategy. This will likely be the use of the existing economizer control. Use this time to build baseline data for each building. During this time, visit each site (March and/or May) and characterize the nominal usage patterns, etc.

June-July, 2001: For each climate type, set up one building with DCV mode and the other with normal economizer mode. (Sacramento sites have Honeywell economizers currently installed.)

August-Fall, 2001: At each climate type, reverse the ventilation control strategies, with one building using DCV and the other set-up for fixed position dampers.

November 2001 – December/January 2002: Maintain the same control strategy for each building for the beginning of the heating season.

December 2001 – February 2002: Reverse ventilation control strategies between the buildings at each climate type. (Do this during a site visit in December 2001 or January 2002.) Change back to the same settings for each room as with the first cooling season phase of June-July, 2001.

<u>RETAIL STORES</u>

<u>The detailed plan for monitoring the retail stores will be finalized after completion of the equipment installation. The plan will likely be as follows.</u>

August-Fall, 2001: After the initial installation and checkout of the control equipment, begin to monitor the buildings at the inland and coastal climate sites with one building in DCV mode and the other using normal economizer control mode.

November 2001 – December/January 2002: Maintain the same control strategy for each building for the beginning of the heating season.

December 2001 – February 2002: Reverse ventilation control strategies between the buildings at each climate type. (Do this during a site visit in December 2001 or January 2002.)

Spring 2002: Reverse the ventilation control strategies from the cooling season data gathered during August and the fall of 2001.

MODELING AND TESTING STRATEGIES FOR EVALUATING VENTILATION LOAD REDUCTION TECHNOLOLGIES

Deliverables 3.1.1a and 4.2.1a

Progress report submitted to:
Architectural Energy Corporation

For the Building Energy Efficiency Program
Sponsored by:
California Energy Commission

Submitted By:

Purdue University

Principal Investigator:	James Braun, Ph.D., P.E.
Research Assistants:	Kevin Mercer
	Tom Lawrence, P.E.

April 2001

Mechanical Engineering
1077 Ray W. Herrick Laboratories
West Lafayette, IN 47907-1077
(765) 496-6008
(765) 494-0787 (fax)

RAY W. HERRICK
LABORATORIES
PURDUE ENGINEERING

TABLE OF CONTENTS

I. INTRODUCTION

A. Scope

The heating and cooling loads associated with ventilation can contribute significantly to the total energy requirements for a commercial space being conditioned. In recent years, several different approaches have been proposed to reduce ventilation loads including enthalpy exchangers, economizers, demand-control ventilation and ventilation heat recovery heat pumps. However, different technologies may be appropriate for different environments and buildings.

This project will focus on identifying appropriate applications and locations for ventilation load reduction technologies within the state of California. The performance of economizer, enthalpy exchanger, demand-controlled ventilation and heat recovery heat pump technologies will be compared for different types of buildings and locations. For demand-controlled ventilation, field sites are being established in coastal and inland sites in both northern and southern California. Three different building types are being considered with two nearly identical buildings for each location so that direct comparisons between the performance of fixed ventilation and demand-controlled ventilation can be made. Data from the field sites will be compared with simulation results in order to validate computer models. The models will then be used to evaluate the cost savings potential for this technology for other buildings and locations. In addition, the models will also consider economizer, enthalpy exchanger, and heat pump heat recovery technologies. The performance of all these technologies will be compared in terms of their cost effectiveness. As a further validation of the simulation results, an additional field will be established for testing the heat pump heat recovery unit.

B. Purpose of this Report

This progress report presents an overview of the modeling approach and input data to be used in evaluating the energy savings associated with each of the ventilation load reduction technologies. In addition, an overview of the preliminary test plan and field site monitoring setup for the heat pump heat recovery unit is given.

II. VENTILATION LOAD REDUCTION TECHNOLOGIES

A. Economizer

An economizer uses outside air to reduce or eliminate the mechanical cooling required to condition a building. This accessory usually includes an outside air damper, a relief damper, a return air damper, filters, an actuator and linkages. An economizer can be installed with any of the other three ventilation energy savings technologies that will be considered in this study. When the outdoor conditions are suitable, the outdoor air dampers switch from their minimum position (minimum ventilation air) to fully open. For a dry-bulb economizer, this switch point occurs when ambient air is less than a specified value. This switch point should be less than the switch point to return to minimum outside air in order to ensure stable control. The economizer switchover temperature may be significantly lower than the return air temperature (e.g., 10 F lower) in humid climates where latent ventilation loads are significant. However, in dry climates, the switchover temperature may be close to the return temperature (e.g., 75 F). An enthalpy (or wet-bulb) economizer compares the outside and return air enthalpies (or wet-bulb temperatures) in order to initiate or terminate economizer operation. In general, enthalpy economizers yield lower energy costs than dry-bulb economizers, but require a humidity measurement. With either economizer, the outside air damper modulates the flow to maintain a mixed air temperature set point, and when this set point can no longer be achieved, the compressor is engaged (Howel et al., 1998).

B. Enthalpy Exchanger

A rotary air-to-air enthalpy exchanger, sometimes called a heat recovery wheel, is a revolving cylinder filled with an air permeable medium with a large internal surface area for contact with the air passing through it. Adjacent supply and exhaust streams each flow through half the exchanger in a counter-flow pattern as illustrated in Figure 1.

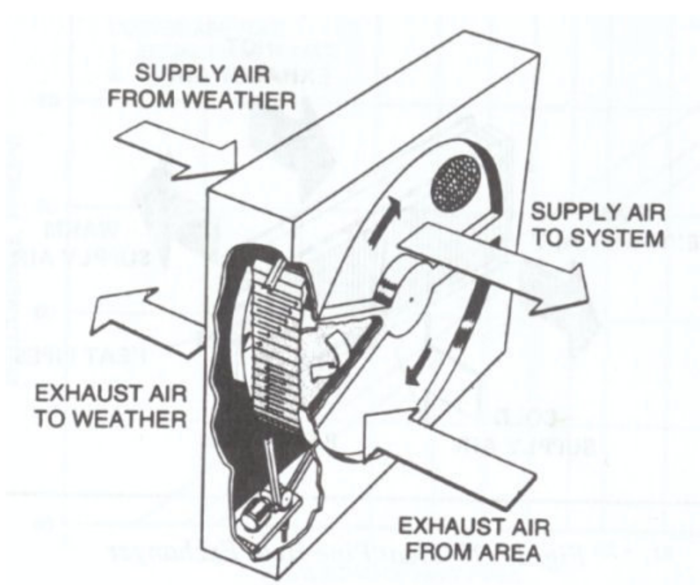


FIGURE 1. FLOW DIRECTION IN AN ENTHALPY EXCHANGER

Sensible heat is recovered as the medium picks up and stores heat from the hot airstream and gives it up to the cold airstream. Latent heat is transferred as the medium condenses moisture from the airstream having the higher humidity ratio, with a simultaneous release of heat. The medium then releases the moisture through evaporation into the airstream with the lower humidity ratio. The enthalpy exchanger medium is fabricated from metal, mineral, or man-made materials and classified as providing either random flow or directionally oriented flow through their structures (Howel et al., 1998). An enthalpy exchanger works for both heating and cooling and can allow for 100% outside air.

C. Demand Controlled Ventilation

The energy requirements to heat or cool a building can be reduced by modulating ventilation air in response to the number of occupants in the building at any given time. This can be accomplished by controlling the ventilation air to maintain a specific CO_2 level within the building. This strategy is referred to as demand-controlled ventilation (DCV). Brandemuehl and Braun (1999) performed a simulation study for a number of different buildings and locations and showed that as much as 20% savings in electrical energy for cooling are possible with demand-controlled ventilation. The savings in heating energy associated with demand-controlled ventilation are generally much larger, but are strongly dependent upon the building type and occupancy schedule. Significantly greater savings are possible for buildings with highly variable occupancy schedules and relatively large internal gains. However, the overall cost effectiveness of DCV has not been evaluated and the savings have not been documented in the field.

D. Ventilation Heat Pump Heat Recovery

Carrier's Energy Recycler® accessory, available for 3 to 12.5 ton rooftop units, introduces a technique to help reduce the total load on the primary HVAC system by outside air pre-treatment. Figure 2 illustrates operation of the Energy Recycler® using some example design cooling conditions. In the cooling season, the Energy Recycler® cools and possibly dehumidifies outside air entering the unit, allowing for larger quantities of outside air. The heat is rejected into the exhaust air from the building. The room air is used to cool the condenser coil and thus allows the condenser to operate at a lower temperature than the ambient. During heating season, the Energy Recycler® operates in reverse as a heat pump to extract heat from the exhaust air and pre-heat the outside air. The application of a ventilation heat pump heat recovery units leads to a lower load on the primary equipment. However, the unit requires energy and the overall economics are not known.

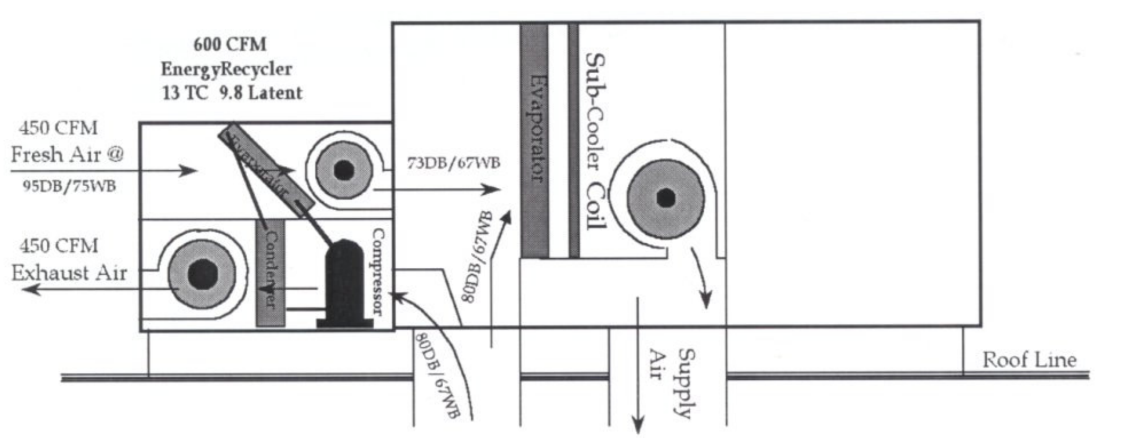


FIGURE 2. ENERGY RECYCLER® SCHEMATIC ATTACHED TO ROOFTOP UNIT

II. SIMULATION APPROACH

The simulations will be performed for a variety of small commercial building types that utilize packaged air conditioning and heating equipment. A computer simulation model is being developed for estimating the energy requirements and life cycle economic impact for the different ventilation load reduction technologies. The model is based upon the tool previously developed by Brandemuehl and Braun (1999). Figure 3 shows a flow diagram of the computer simulation model to be implemented for evaluating these different methods.

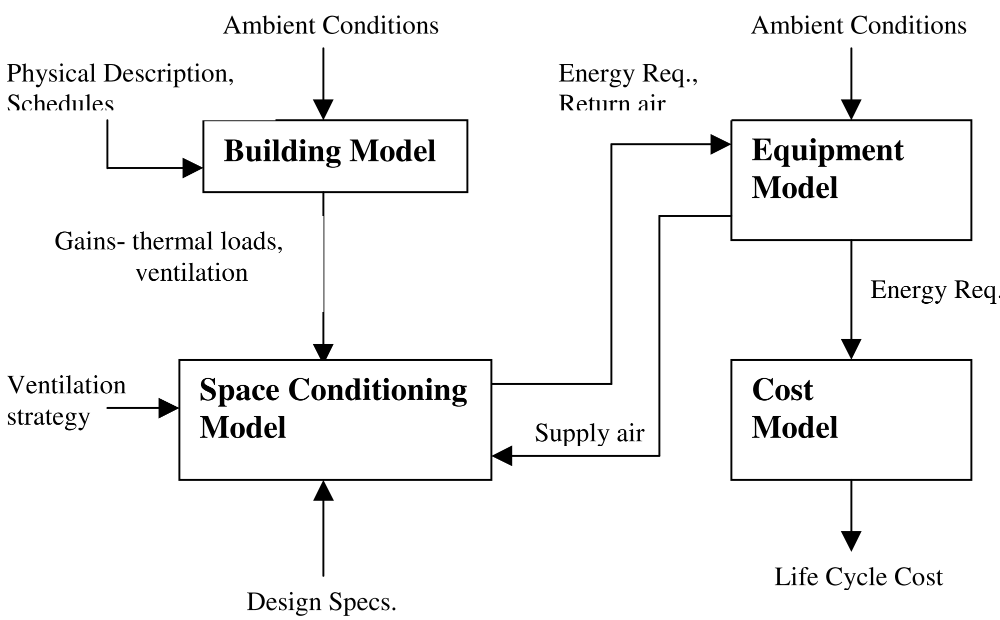


FIGURE 3. FLOW DIAGRAM OF MODELING APPROACH

The model will calculate hourly energy requirements for a particular building type and system type and then use this data to determine the total cost of HVAC operation. The building model will predict the thermal gains to or from the zone based upon transient heat transfer from outside walls and internal sources. The space-conditioning model will solve mass and energy balances for the zone air and then determine return air conditions for the equipment model. The zone air humidity, dry-bulb temperature, and CO2 concentration will be calculated at each hour within the space conditioning model. The ventilation and return air will be mixed according to the ventilation technique being analyzed. The equipment model will use mixed air conditions and the sensible cooling requirement to determine the average supply air conditions. These entering mixed air conditions and supply air conditions will be determined iteratively using a nonlinear equation solver. The energy used by the equipment model will be calculated and used as an input in determining the life cycle cost for each system.

The cost model will incorporate current electricity rates in California and equipment costs to estimate the life cycle cost of the HVAC system for each ventilation control technique. From this economic data, comparisons can be made between all the different combinations of location and building type. The length of the economic analysis will be varied to reflect different potential decision makers.

The nonlinear equation solver to be used in this study is an HVAC building/ energy simulation program called TRNSYS (1996), developed by the Solar Energy Laboratory at the University of Wisconsin-Madison. TRNSYS is a transient systems simulation program with a modular structure. It recognizes a system description language in which the user specifies the components that constitute the system and the manner in which they are connected. The TRNSYS library includes many of the components commonly found in thermal energy systems, as well as component routines to handle input of weather data. The modular structure of TRNSYS gives the program tremendous flexibility and facilitates the addition to the program of mathematical models that are not included in the standard TRNSYS library. An electronic simulation of the previously mentioned ventilation control strategies can thus be added to the TRNSYS library. With this computer simulation in place, several different combinations of location and building type can be simulated to evaluate the performance of all ventilation control strategies.

A. Building Model

The TYPE 56: "Multi-Zone Building" component from the TRNSYS library will be used for the building model. This component models the thermal behavior of a building having up to 25 thermal zones. This is a very detailed model of a building that is built up from individual descriptions of wall layers, windows, internal gain schedules, etc. The model solves individual transient conduction through walls and considers long-wave radiation exchanges within the space. Model inputs include separate hourly heating and cooling setpoints and the model outputs the required heating or cooling rates necessary to maintain the setpoints.

B. Space-Conditioning Model

The space-conditioning model determines return air conditions for the equipment model. The zone sensible heat gain or loss and the specified zone temperature setpoint determines the required average supply air temperature. Given the supply airflow rate and the supply air temperature, the thermal load requirements for the equipment model are determined by the mixed air conditions. These mixed air conditions depend on the ventilation control strategy implemented.

When the DCV control strategy is enabled, a minimum flow rate of ventilation air is determined that will keep the CO_2 concentration in the zone at or below a specified level (Brandemuehl and Braun, 1999). In the absence of DCV, ventilation percentages are based on design conditions for each specific building type from the ASHRAE Standard 62-1999. Table 1 shows the parameters used to estimate the minimum ventilation rates according to building type.

TABLE 1. ASHRAE MINIMUM VENTILATION REQUIREMENTS

Parameter	***Office***	***Retail***	***School***	***Restaurant***	***Hotel***	***Super-market***
Minimum Ventilation per Person, cfm	20	10*	15	20	15	15
Maximum Design Occupancy for minimum ventilation flow, P/1000 ft^2	7	20	50	70	30	8

*Retail store minimum ventilation is based upon an average of 0.25 cfm/ft^2 for upper and lower floors.

For known ventilation flow, zone temperature, and ambient conditions, steady-state mass and energy balances will be applied to the zone and air distribution system to determine average values over each timestep for the return and zone air CO_2 concentration and humidity ratio. These calculations will be based on a fully-mixed zone model, modified by an air exchange effectiveness to account for partial short-circuiting of the supply air to the ceiling return.

Within the TRNSYS environment, the space-conditioning model will be a custom TYPE component that will interact with the TYPE 56 model through inputs and outputs

C. Equipment Model

Packaged rooftop air conditioner with on/off controls will be simulated in this study. The model will use the return air and ambient air conditions to determine the average supply air conditions for the space-conditioning model. The analysis will include air conditioners with gas furnaces and electric auxiliary heat. The supply fan will be on during all hours of occupancy, and the compressor or heater will cycle on and off as necessary to maintain the zone temperature at its set point. Models for a direct expansion air conditioner will taken from the ASHRAE Secondary Toolkit (Brandemuehl, et al., 1993) and adapted for this project. The secondary toolkit contains a library of subroutines and functions that have been debugged and documented. The direct expansion and heat pump models are based upon correlations used in DOE 2.1E. These models estimate capacity (cooling or heating) and power consumption as a function of mixed air and ambient conditions for typical devices. The outputs are scaled according to capacity and efficiency values that are specified for ARI rating conditions. Both high and moderate efficiency units will be considered in this study. For cooling, both sensible and total cooling capacities are determined. Iteration with the space-conditioning model is required, since the space humidity level is determined by the moisture removal rate of the equipment, which is affected by the mixed air humidity.

Models for a heat pump will also be taken from the ASHRAE Secondary Toolkit and adapted for modeling the heat pump heat recovery unit. Laboratory test data will be taken over a wide range of conditions and used to adjust coefficients of the model.

D. Cost Model

The cost model will consider utility and initial equipment costs to determine life-cycle costs (including inflation, alternative investments, taxes, financing, depreciation, maintenance, etc.). Utility rate information will be gathered for each location considered, including energy and demand rates. The life-cycle costs for different ventilation load technologies will be compared leading to an overall assessment.

III. SIMULATION INPUT DATA

A. Selected Locations

TMY2 (NREL, 1995) data for a number of locations in and near California will be used in the simulation studies. The National Renewable Energy Laboratory, NREL, has extracted data from the National Solar Research Data Base, NSRDB, for the years of 1961 to 1990 to produce the Typical Meteorological Year, or TMY weather data. TMY data is a set of hourly values of solar radiation and meteorological elements for a one-year period. It consists of months selected from individual years and concatenated to form a complete year. TMY2 data is a more recent version that was completed in March of 1994. Two minor errors that affected about 10% of the original TMY data stations were corrected in this version.

For this study, locations were selected from the available TMY2 data that are representative of diverse climates acrorss California. The selected cities are shown in Table 2.

TABLE 2. CALIFORNIA AND NEVADA CITIES FOR TRNSYS SIMULATIONS

City	Latitude Deg	Latitude Min	Longitude Deg	Longitude Min	Elev. (m)
Arcata	N40	59	W124	06	69
Bakersfield	N35	25	W119	03	150
Daggett	N34	52	W116	47	588
Fresno	N36	46	W119	43	100
Los Angeles	N33	56	W118	24	32
Sacramento	N38	31	W121	30	8
San Diego	N32	44	W117	10	9
San Francisco	N37	37	W122	23	5
Santa Maria	N34	54	W120	27	72
Reno, NV	N39	30	W119	47	1341
Las Vegas, NV	N36	05	W115	10	664

Arcata, San Francisco, Santa Maria, Los Angeles and San Diego are on the west coast of California proceeding from the north to south. These areas have very temperate climates averaging around 80°F and 40 to 50% relative humidity during the summer season. During winter months, the mean temperature drops to the low 40's and perhaps on occasion the upper 30's. Sacramento, Fresno, Bakersfield, and Baggett are inland from the west coast, approximately in the middle of the state. These areas are much hotter in the summer season, especially Bakersfield and Baggett. Las Vegas and Reno, Nevada, were both chosen to represent the eastern border area of California. Las Vegas temperatures range from the 20's during the winter and above 100°F during the summer. Climates near Reno are in the high 90's during the summer and lower teens in the winter.

B. Buildings

Brandemuehl and Braun (1999) considered four different types of buildings in their study: office, large retail store, school, and sit-down restaurant. Descriptions for these buildings were obtained from prototypical descriptions of commercial buildings developed by Lawrence Berkeley National Laboratory (Huang and Franconi, 1995). Table 3 gives data to describe these buildings. The current study will expand upon the previous work in that the cost effectiveness of DCV and other ventilation load reduction technologies will be considered and compared.

Table 3: Prototypical Building Characteristics use by Brandemuehl and Braun (1999)

Characteristic	Office	Large Retail	School	Sit-Down Restrnt.
Floor area (ft^2)	6600	80,000	9,600	5250
Floors	1	2	2	1
Percent glass	15	15	18	15
Window R-value	1.6	1.7	1.7	1.5
Window shading coeff.	0.75	0.76	0.73	0.80
Wall R-value	5.6	4.8	5.7	4.9
Roof R-value	12.6	12.0	13.3	13.2
Wall material	Masonry	Masonry	Masonry	Masonry
Roof material	Built-up	Built-up	Built-up	Built-up
Weekday hours (hrs/day)	11	14	Varies	17
Weekend hours (hrs/day)	5	14	Varies	17
Equipment power (W/ft^2)	0.5	0.4	0.8	2.0
Lighting power (W/ft^2)	1.7	1.6	1.8	2.1

Four additional building types from the LBL report will be considered in the current study: small retail stores, hotels, supermarkets, and middle schools. Tables 4, 5, 6 and 7 give data that describe these buildings. All of the simulated buildings will utilize packaged air conditioning equipment with a natural gas electric heater. For supermarkets, both old and new buildings will be simulated. The construction of this building type has changed dramatically over the last 30 years. However, many older buildings still are in commission and could be retrofit with ventilation load reduction technologies.

The LBL study consulted the 1989 CBECS (EIA, 1992) to determine total floor area for each building type, vintage, and climatic zone, the percentages of floor area heated or cooled, and the total energy use of the building type. The building shell characteristics and schedules were derived from the LBL study; however, the LBL study derived the data from a previous study conducted by (Huang et al., 1990) along with updates from the 1989 CBECS.

In addition to the buildings from the LBL study, the field site buildings will also be simulated. Site-specific data necessary for simulating system performance is currently being gathered (see report on the Description of Field Sites for Deliverables 2.1.1a and 3.1.1a). Once all data has been gathered from the field sites, this information will serve to validate the computer simulation model before any HVAC simulations are conducted for other buildings and locations. For DCV, the field sites have been chosen with two nearly identical buildings for each site. This will allow some degree of side-by-side testing for comparison of fixed minimum ventilation and DCV. However, more importantly, the test data will be used for validating the models and the predicted savings. Then, the improved models can be used to evaluate savings for the other technologies and locations.

TABLE 4. CHARACTERISTICS OF A MODELED SMALL RETAIL STORE

	Parameters
FLOOR-AREA	
Building area (ft^2)	6400
Floors	1
SHELL	
Percent Glass	15
Window R-value	1.67
Window shading co-efficient	0.84
Wall R-value	4.83
Roof R-value	12.04
Wall material	masonry
Roof material	built-up
OCCUPANCY	
Occupancy (ft^2/pers)	1635
Weekday hours (hrs/day)	12
Weekend hours (hrs/day)	4
EQUIPMENT	
Power density (W/ft^2)	0.50
Full Eqp hours (hrs/yr)	3480
LIGHTING	
Power density (W/ft^2)	1.7
Full lighting hours (hrs/yr)	4412
SYSTEM AND PLANT CHARACTERISTICS	
System type	Packaged single-zone w/ economizer
Heating plant	Gas furnace
Cooling plant	Direct expansion

TABLE 5. CHARACTERISTICS OF MODELED HOTEL PROTOTYPES

	Large hotels	Small hotels (Motels)
FLOOR-AREA		
Building area (ft^2)	250000	12000
Floors	10	2
SHELL		
Percent Glass	35	21
Window R-value	1.67	1.71
Window shading co-efficient	0.74	0.76
Wall R-value	6.16	5.32
Roof R-value	14.00	13.16
Wall material	masonry	masonry
Roof material	built-up	shingle/ siding
OCCUPANCY		
Occupancy (ft^2/pers)	210	120
Weekday hours (hrs/day)	24	24
Weekend hours (hrs/day)	24	24
EQUIPMENT		
Power density (W/ft^2)	0.72	0.69
Full Eqp hours (hrs/yr)	2722	2826
LIGHTING		
Power density (W/ft^2)	1.18	1.06
Full lighting hours (hrs/yr)	5157	3443
SYSTEM AND PLANT CHARACTERISTICS		
System type	Packaged single-zone w/ economizer	Packaged single-zone w/ economizer
Heating plant	Gas furnace	Gas furnace
Cooling plant	Direct expansion	Direct expansion

TABLE 6. CHARACTERISTICS OF MODELED SUPER-MARKETS

	Supermarket	
	old	new
FLOOR-AREA		
Building area (ft^2)	21300	21300
Floors	1	1
SHELL		
Percent Glass	15	15
Window R-value	1.51	1.60
Window shading co-efficient	0.82	0.79
Wall R-value	3.3	5.8
Roof R-value	9.2	11.8
Wall material	masonry	masonry
Roof material	shingle/ siding	shingle/ siding
OCCUPANCY		
Occupancy (ft^2/pers)	227	227
Weekday hours (hrs/day)	18	18
Weekend hours (hrs/day)	18	18
EQUIPMENT		
Power density (W/ft^2)	1.20	1.20
Full Eqp hours (hrs/yr)	5168	5168
LIGHTING		
Power density (W/ft^2)	2.4	2.4
Full lighting hours (hrs/yr)	7816	7816
SYSTEM AND PLANT CHARACTERISTICS		
Numer of systems	5 (office, storage, deli, bakery, sales)	
System type	Constant-vol. single-zone	Variable-air vol. single-zone
Heating plant	Gas furnace	
Cooling plant	Direct expansion	

TABLE 7. CHARACTERISTICS OF MODELED MIDDLE SCHOOL PROTOTYPE

	Parameters
FLOOR-AREA	
Building area (ft^2)	136000
Floors	1
SHELL	
Percent Glass	6
Window R-value	1.39
Window shading co-efficient	0.85
Wall R-value	2.38
Roof R-value	7.56
Wall material	masonry
Roof material	metal surface
OCCUPANCY	
Occupancy (ft^2/pers)	2085
Weekday hours (hrs/day)	12
Weekend hours (hrs/day)	4
EQUIPMENT	
Power density (W/ft^2)	0.30
Full Eqp hours (hrs/yr)	6462
LIGHTING	
Power density (W/ft^2)	0.8
Full lighting hours (hrs/yr)	3638
SYSTEM AND PLANT CHARACTERISTICS	
System type	Packaged single-zone w/ economizer
Heating plant	Gas furnace
Cooling plant	Direct expansion

IV. TESTING

A. Overview

Two distinct types of testing will be conducted for the DCV and ventilation heat pump heat recovery projects. First of all, the Carrier heat pump heat recovery unit will be tested in the laboratory over a wide range of conditions to be encountered in the field. These data will be used to build performance maps for the unit that will be integrated in the simulation tool. Secondly, field tests will be performed for DCV and heat pump heat recovery. An overview of the data flow for the testing and evaluation phase of these projects is given in Figure 4.

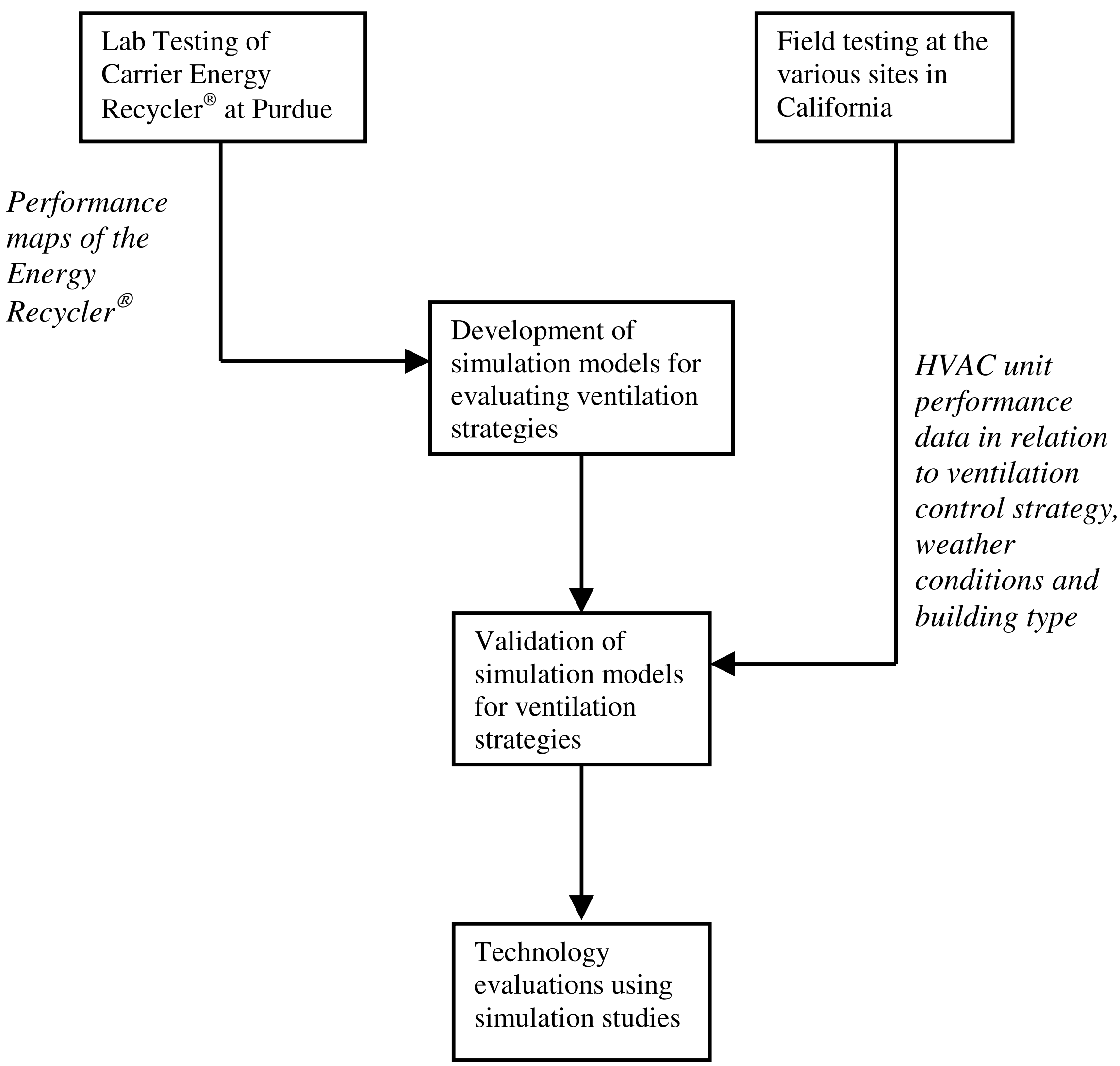


FIGURE 4. LAB AND FIELD TESTING DATA FLOW

B. Lab Testing of the Carrier Energy Recycler® Heat Pump

Project 4.2 is intended to demonstrate the savings potential for application of ventilation recovery heat pumps. This will be done primarily using simulation studies for various building and climate types found throughout California. To develop the simulation model, it is necessary to have accurate performance data for the ventilation recovery heat pump. Therefore, the first phase of Project 4.2 will focus on laboratory testing of a representative unit from Carrier. The environmental chambers at the Ray W. Herrick Laboratories will be used for this testing.

Carrier Corporation, as a sponsor of this program, has provided one of their Energy Recycler® units. This same unit will be used for both laboratory testing and field testing. The unit size was selected based on a field test site at a school that utilizes a Carrier 6-ton rooftop unit with gas heating. This unit was shipped to Purdue in late February of 2001. (See the photo in Figure 5.)

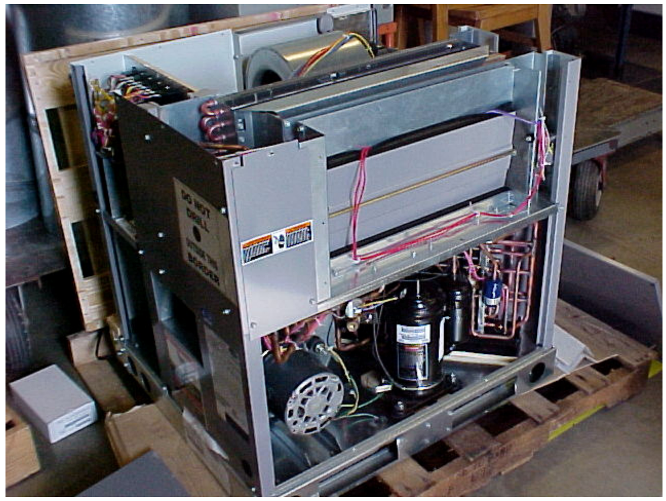


FIGURE 5. CARRIER ENERGY RECYCLER® HEAT PUMP AT HERRICK LAB
(SIDES REMOVED FOR CLARITY)

The ventilation heat pump is scheduled for testing at Herrick Laboratory beginning in May of 2001. The testing will result in a performance map of the unit that covers the complete expected operating envelope for the ambient and return air states. The expected range of the operating conditions for cooling and heating mode testing are given in Table 8. It is only necessary to vary humidity for the evaporator air stream (outside air for cooling mode and return air for heating mode) since performance is relatively independent of humidity when moisture is not condensed.

TABLE 8. OPERATING ENVELOPE FOR LAB TESTING OF THE CARRIER ENERGY RECYCLER® HEAT PUMP

Cooling Mode	
Ambient Temperature	50° to 120° F
Ambient Humidity	10% to 100%
Return Air Temperature	55° to 90° F
Return Air Humidity	Not varied
Heating Mode	
Ambient Temperature	-10° to 55° F
Ambient Humidity	Not varied
Return Air Temperature	50° to 80° F
Return Air Humidity	30% to 80%

The model will correlate sensible and total cooling capacity and power consumption as a function of the entering states and flow rates. The model will then be incorporated into the system model.

C. Field Testing

Field test data will be gathered at a total of 13 different sites in California: Twelve of the test sites are the ones being set up for joint evaluation of the demand controlled ventilation and the gathering of data for field evaluation of the fault detection and diagnostics algorithms. A detailed discussion of these sites and the test plan is included in the separate report: "Description of Field Test Sites".

The 13th site is for the heat pump heat recovery project. This site will be at one of the school districts (Woodland Joint Unified) where the modular schoolrooms are being monitored for DCV. The site selected is at the Junior High School for this district, and it has a 6-ton Carrier rooftop unit with gas heating.

The field testing for the ventilation recovery heat pump will involve two phases. The first phase, initiated in March 2001, was to install a Virtual Mechanic monitoring system on the existing rooftop unit at the California site. Performance data on this unit and the conditioned space will be collected for use in developing a baseline for the unit before installation of the Energy Recycler®. Once the laboratory testing is completed, the heat pump will be installed in the field and the second phase of the field testing initiated. It is anticipated that the field installation will occur during the July-August of 2001 time frame.

Table 9 gives a detailed list of the field test data for the Energy Recycler® as it will be set up for baseline data gathering. Additional sensors will be added when the Energy Recycler® is installed this summer. Detailed lists of test data for the other twelve field sites is contained in the Purdue report titled "Description of Field Test Sites".

TABLE 9. DATA LIST FOR FIELD TESTING OF THE VENTILATION RECOVERY HEAT PUMP

Channel #	Data Point
SENSOR CHANNELS	
Power Transducer Channels	
1	Unit voltage, L1
2	Unit voltage, L2
3	Unit voltage, L3
4	Unit total current, L1
5	Not Used
6	Unit total current, L3
Other Analog Input Data	
7	SPARE - (Use later with heat pump)
8	SPARE - (Use later with heat pump)
9	SPARE - (Use later with heat pump)
10	SPARE - (Use later with heat pump)
11	SPARE - (Use later with heat pump)
12	SPARE - (Use later with heat pump)
13	SPARE - (Use later with heat pump)
14	SPARE - (Use later with heat pump)
15	Mixed air temperature
16	Return air temperature
17	Supply air temperature, before heater
18	Supply air temperature, after heater
19	Condenser inlet air temperature
20	Condenser outlet air temperature
21	Suction line temperature, rooftop unit
22	Discharge line temperature, rooftop unit
23	SPARE - (Use later with heat pump)
24	SPARE - (Use later with heat pump)
25	Evaporation temperature, rooftop unit
26	Condensation temperature, rooftop unit
27	Outdoor air temperature
28	Outdoor air humidity
29	Building zone temperature A
30	Building zone temperature B
31	Building zone temperature C
32	Building zone temperature D
CALCULATED DATA CHANNELS	
33-50	NOT USED
51-56	NOT USED

TABLE 9. DATA LIST FOR FIELD TESTING OF THE VENTILATION RECOVERY HEAT PUMP (CONT'D)

Channel	***Data Point***
57	superheat, stage 1
58	subcooling, stage 1
59	evaporating temperature, stage 1
60	condensing temperature, stage 1
61	condensing temperature over ambient (CT-AIC), stage 1
62	NOT USED
63	NOT USED
64	NOT USED
65	NOT USED
66	NOT USED
67	evaporator temperature difference (RA-SA)
68	NOT USED
69	NOT USED
70	unit power (kW)
71	unit KWh
72	unit MWh
73	compressor 1 power (kW)
74	compressor 1 KWh
75	compressor 1 MWh
76	compressor Vent Heat Pump Unit power (kW)
77	compressor Vent Heat Pump Unit KWh
78	compressor Vent Heat Pump Unit MWh
79	digital input 1, supply fan, run time (8 hours)
80	digital input 1, supply fan, run time (seconds)
81	digital input 2, cooling 1, run time (8 hours)
82	digital input 2, cooling 1, run time (seconds)
83	digital input 3, cooling Vent HP, run time (8 hours)
84	digital input 3, cooling Vent HP, run time (seconds)
85	digital input 4, heat 1, run time (8 hours)
86	digital input 4, heat 1, run time (seconds)
87	digital input 5, heat Vent Heat pump, run time (8 hours)
88	digital input 5, heat Vent Heat pump, run time (seconds)
89	digital input 6 run time (8 hours)
90	digital input 6 run time (seconds)
91	time since reset accumulators (8 hours)
92	time since reset accumulators (seconds)
93	up time (8 hours)
94	up time (seconds)
95	board temperature (F)
96	board battery voltage (V)

TABLE 9. DATA LIST FOR FIELD TESTING OF THE VENTILATION RECOVERY HEAT PUMP (CONT'D)

Digital Channels

1	Supply fan contact (fan om / fan off)
2	Low voltage control signal for compressor main unit
3	Low voltage control signal for compressor, heat pump
4	Heating mode signal
5	
6	

VII. REFERENCES

ASHRAE. 1999. "ANSI/ASHRAE Standard 62-1999, "Ventilation for Acceptable Indoor Air Quality", Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Brandemuehl, M.J., Gabel, S., and Andresen, I. 1993. HVAC2 Toolkit: Algorithms and Subroutines for Secondary HVAC System Energy Calculations. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.

Brandemuehl, M., and Braun, J. 1999. "The Impact of Demand-Controlled and Economizer Ventilation Strategies on Energy Use in Buildings", Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Carrier Corporation. 1999. "Product Data: 62AQ Energy$Recycler Energy Recovery Accessory For 3 to 12.5 Ton Rooftops", Form 62AQ-1PD. Syracuse, New York.

Energy Information Administration (EIA) 1992. "Commercial Building Energy Consumption and Expenditures", U.S. Department of Energy, Washington, D.C.

Howel, Ronald H., Sauer, Harry J., and Coad, William J. 1998. *Principles of Heating, Ventilating and Air Conditioning*, Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Huang, Y.J., Akbari, H., Rainer, L., and Ritschard, R.L. 1990. "481 Prototypical Commercial Buildings for Twenty Urban Market Areas (Technical documentation of building loads data base developed for the GRI Cogeneration Market Assessment Model)", LBL Report 29798, Lawrence Berkley National Laboratory, Berkley CA.

Huang, Y.J., and Franconi, E. 1995. "Commercial Heating and Cooling Loads Component Analysis", LBL Report 37208, Lawrence Berkley National Laboratory, Berkley CA.

National Renewable Energy Laboratory (NREL) 1995. "User's Manual for TMY2s: Typical Meteorological Years", U.S. Department of Energy. Golden, Colorado.

Solar Energy Laboratory (SEL) 1996. "TRNSYS: A Transient System Simulation Program User's Manual", University of Wisconsin. Madison, Wisconsin.

NISTIR 6729

State-of-the-Art Review of CO_2 Demand Controlled Ventilation Technology and Application

Steven J. Emmerich

Andrew K. Persily

NIST CENTENNIAL 1901-2001

NIST

National Institute of Standards and Technology

Technology Administration, U.S. Department of Commerce

NISTIR 6729

State-of-the-Art Review of CO_2 Demand Controlled Ventilation Technology and Application

Steven J. Emmerich
Andrew K. Persily
Building and Fire Research Laboratory

Prepared for:
Architectural Energy Corporation
Boulder, Colorado

March 2001

U.S. Department of Commerce
Donald L. Evans, Secretary

National Institute of Standards and Technology
Karen H. Brown, Acting Director

ABSTRACT

The control of outdoor air intake rates in mechanically ventilated buildings based on indoor carbon dioxide (CO_2) levels, often referred to as CO_2 demand controlled ventilation (DCV), has the potential for reducing the energy consumption associated with building ventilation in some commercial and institutional buildings. Carbon dioxide DCV has been discussed, promoted, studied and demonstrated for about twenty years, but questions still remain regarding the actual energy savings potential as a function of climate, ventilation system features, and building occupancy. In addition, questions exist as to the indoor air quality (IAQ) impacts of the approach and the best way to implement CO_2 DCV in general and in a given building. This report presents a state-of-the-art review of CO_2 DCV technology and application including discussion of the concept and its application, and a literature review. In addition the regulatory and standard requirements impacting CO_2 DCV are also examined.

Keywords: carbon dioxide, control, energy efficiency, indoor air quality, ventilation

TABLE OF CONTENTS

1. INTRODUCTION

Many ventilation requirements and recommendations, e.g., ASHRAE Standard 62-1999, are in the form of outdoor airflow rates per person expressed as L/s or cfm per person. Mechanical ventilation systems are therefore designed to provide a minimum level of outdoor air based on the designed occupancy level multiplied by the per-person ventilation requirement. However, a space that is occupied at less than its design level may still be ventilated at this design minimum rate, often resulting in increased energy consumption beyond that which would be required based on the actual occupancy. Furthermore, early during a given day of building occupancy, contaminants generated by people and their activities will not yet have reached their ultimate levels based on the transient nature of the situation. As a result, it is sometimes possible to delay or lag the onset of the design ventilation rate to take credit for this transient effect. A number of approaches have been proposed to account for actual occupancy levels and to provide the ventilation rate corresponding to the actual rather than design occupancy. These include time-based scheduling when the occupancy patterns are well known and predictable, occupancy sensors to determine when people have entered a space (though not necessarily how many) and CO_2 sensing and control as a means of estimating the number of people in a space or at least the strength of occupant-related contaminant sources.

Controlling outdoor air intake rates using CO_2 demand controlled ventilation (DCV) offers the possibility of reducing the energy penalty of over-ventilation during periods of low occupancy, while still ensuring adequate levels of outdoor air ventilation. As discussed later in this report, depending on climate and occupancy patterns, CO_2 DCV may provide significant energy savings in commercial and institutional buildings. While a number of studies have suggested the extent of such savings via field studies and computer simulations, additional work is needed to better define the magnitude of energy savings possible and the dependence of these savings on climate, building and system type, control approach, and occupancy patterns. In addition, important issues remain to be resolved in the application of CO_2 DCV including how best to apply the control approach, including issues such as which control approach to use in a given building, sensor location, sensor maintenance and calibration, and the amount of baseline ventilation required to control contaminant sources that don't depend on the number of occupants.

This report presents a state-of-the-art review of CO_2 DCV technology and its application in commercial and institutional buildings. Following this introduction, the next section presents the theoretical background of CO_2 DCV including discussions of CO_2 generation rates by people, the relationship of indoor CO_2 to building ventilation rates, and the basic concept of controlling ventilation based on indoor CO_2 levels. The third section of the report is a literature review of previous research on CO_2 DCV, including field demonstration projects, computer simulation studies, studies of sensor performance and location, and discussions of the application of the approach. The next section of the report contains an update on CO_2 sensor technology as it applies to DCV. The manner in which CO_2 DCV is addressed in standards and other regulations, including California's Energy Efficiency Standards (often referred to as Title 24), is presented in section five of this report. The report also contains two appendices. The first appendix discusses how CO_2 DCV relates to the four issues identified by the California Energy Commission Public Interest Energy Research Request for Proposal #400-99-401 that resulted in this project. The second appendix summarizes preliminary guidance on the application of CO_2 DCV based on the material reviewed in preparing this report. While future phases of this effort are intended to develop more definitive guidance, this appendix attempts to capture some of the guidance that has been developed to date while identifying some issues that remain to be resolved.

2. TECHNICAL BACKGROUND: CARBON DIOXIDE IN BUILDINGS

In order to evaluate the possibilities and application of CO_2 DCV, it is important to understand the dynamics of indoor CO_2. This section discusses these dynamics, including indoor CO_2 generation rates, how indoor CO_2 levels relate to ventilation, and how CO_2 can be used to control ventilation. Much of this material is based on an earlier publication by Persily (1997). This discussion does not cover the issue of using indoor CO_2 to measure or estimate building ventilation rates, but rather is focused on issues related to ventilation rate control based on indoor CO_2 levels. Persily (1997) contains a discussion of the measurement issue, as does ASTM Standard D6245 (1998).

2.1 Carbon Dioxide Generation Rates

While it is not critical to the application of CO_2 DCV, the emission rate of occupant generated CO_2 is certainly a relevant issue in this discussion. This section discusses the rate at which people generate CO_2.

People generate CO_2, and consume oxygen, at a rate that depends primarily on their body size and their level of physical activity. The relationship between activity level and the rates of carbon dioxide generation and oxygen consumption is discussed in the ASHRAE Fundamentals Handbook (ASHRAE 1997). The rate of oxygen consumption V_{O2}, in L/s, of a person is given by the following equation

$$V_{O2} = \frac{0.00276 A_D M}{(0.23RQ + 0.77)} \tag{1}$$

When using inch-pound units, V_{O2} is expressed in cfm and Equation (1) takes the form

$$V_{O2} = \frac{0.000543 A_D M}{(0.23RQ + 0.77)} \tag{2}$$

where RQ is the respiratory quotient, i.e., the relative volumetric rates of carbon dioxide produced to oxygen consumed. M is the level of physical activity, or the metabolic rate per unit of surface area, in mets (1 met = 58.2 W/m^2 = 18.5 Btu/h·ft^2). A_D is the DuBois surface area in m^2, which can be estimated by the following equation

$$A_D = 0.203 H^{0.725} W^{0.425} \tag{3}$$

where H is the body height in m and W is the body mass in kg. When using inch-pound units, A_D is in ft^2, 0.203 is replaced with 0.660, H is in ft and W is in lb. For an average size adult, A_D equals about 1.8 m^2 (19 ft^2). Additional information on body surface area is available in the EPA Exposure Factors Handbook (EPA 1999). The value of RQ depends on diet, the level of physical activity and the physical condition of the person. It is equal to 0.83 for an average size adult engaged in light or sedentary activities. RQ increases to a value of about 1 for heavy physical activity, about 5 met. Given the expected range of RQ, it has only a secondary effect on carbon dioxide generation rates.

The carbon dioxide generation rate of an individual is therefore equal to V_{O2} multiplied by RQ. Figure 1 shows oxygen consumption and carbon dioxide generation rates as a function of physical activity for an average sized adult with a surface area of 1.8 m^2 (19 ft^2) and RQ = 0.83. Based on Equation 1, the carbon dioxide generation rate corresponding to an average size adult engaged in office work (1.2 met) is about 0.0052 L/s (0.011 cfm). However, the generation rate depends strongly on activity level and can cover a range from less than 0.0050 L/s (0.011 cfm) at 1 met to as high as 0.010 L/s (0.021 cfm) at about 2 met for the occupants of an office building.

The carbon dioxide generation rate for a child with $A_D = 1\ m^2$ ($11\ ft^2$) and a physical activity level of 1.2 met is equal to 0.0029 L/s (0.0061 cfm). When making calculations that use the carbon dioxide generation rate in a building, one must consider the level of physical activity and the size of the building occupants. Chapter 8 of the ASHRAE Fundamentals Handbook, Thermal Comfort, (ASHRAE 1997) contains typical met levels for a variety of activities. Some of these values are reproduced in Table 1.

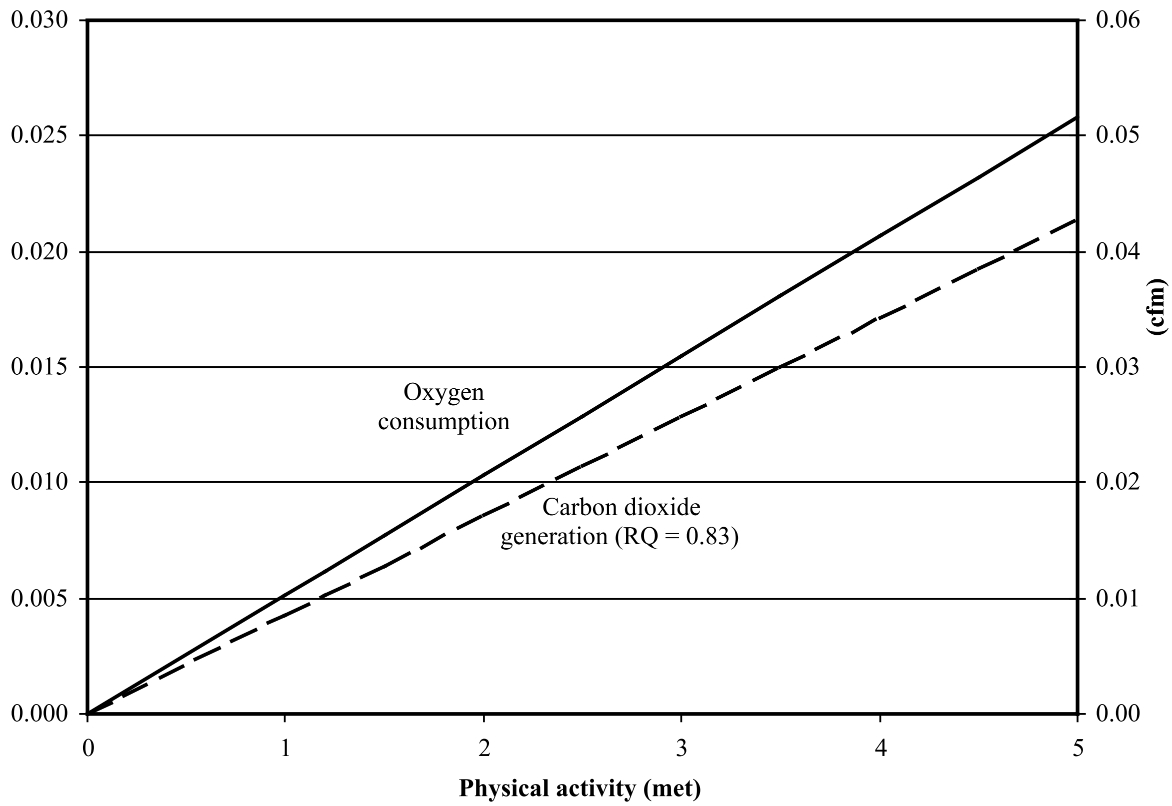


Figure 1 CO_2 generation and O_2 consumption as a function of physical activity (for an average size adult)

Oxygen depletion is sometimes cited as a cause of indoor air quality complaints in buildings. Based on the oxygen consumption rates determined with Equation 1, O_2 depletion due to low ventilation rates is almost never an issue. Given an activity level corresponding to office work, about 1.2 met, the oxygen consumption rate of an individual equals 0.006 L/s (0.013 cfm). At an outdoor air ventilation rate of 7.5 L/s (16 cfm) per person, the steady-state indoor oxygen concentration is reduced from its typical outdoor level of 21 % to 20.9 %. At 2.5 L/s (5.3 cfm), the indoor oxygen concentration is reduced to 20.8 %. Reduced oxygen concentrations do not affect human health or comfort until oxygen levels decrease below 19.5 % (NIOSH 1987), which corresponds to an outdoor air ventilation rate of 0.4 L/s (0.8 cfm) per person. Such low oxygen concentrations are not typically encountered indoors, except in confined spaces where another gas is displacing oxygen or during fires.

Activity	Met
Seated, quiet	1.0
Reading and writing, seated	1.0
Typing	1.1
Filing, seated	1.2
Filing, standing	1.4
Walking at 0.9 m/s (2 mph)	2.0
House cleaning	2.0-3.4
Exercise	3.0-4.0

Table 1 Typical Met Levels for Various Activities (ASHRE 1997)

2.2 Carbon Dioxide, Indoor Air Quality and Ventilation

There has been a great deal of confusion over the years with respect to the relationship of indoor CO_2 levels, indoor air quality and ventilation (Persily 1993 and 1997), much of which has carried over into the discussion of demand controlled ventilation. One of the primary issues has been the significance of indoor CO_2 levels as an indicator of indoor air quality and the ability to maintain acceptable indoor air quality based on the control of indoor CO_2 levels. This section discusses the significance of indoor CO_2 levels in the context of indoor air quality and ventilation.

Indoor CO_2 concentrations have been referred to as an indicator of indoor air quality, often without describing the specific association between carbon dioxide and indoor air quality that is being indicated. There are a number of relationships that could be implied in discussing carbon dioxide and indoor air quality including the relationship between carbon dioxide concentrations and occupant perceptions of the indoor environment, the relationship between carbon dioxide concentrations and the concentrations of other indoor contaminants, and the relationship between carbon dioxide and outdoor air ventilation rates. While some of these relationships are relatively well understood, and in some cases well founded, others have not been documented experimentally or theoretically. In other words, indoor carbon dioxide concentrations can be used to indicate specific and limited aspects of indoor air quality, but do not provide an overall indication of the quality of indoor air. However, this relationship is almost an entirely different issue from that of controlling outdoor air intake rates based on CO_2 levels as discussed below.

Carbon Dioxide and Indoor Air Quality

Carbon dioxide is not generally considered to be a health concern at typical indoor concentrations. The time-weighted average threshold limit value (8 h exposure and a 40 h work week) for carbon dioxide is 9000 mg/m^3 (5000 ppm(v)), and the short-term exposure limit (15 min exposure) is 54 000 mg/m^3 (30 000 ppm(v)) (ACGIH 2001). A number of studies at elevated concentrations, about 5 % carbon dioxide in air or 90 000 mg/m^3 (50 000 ppm(v)), have been performed, and the lowest level at which effects have been seen in humans and animals is about 1 %, i.e., 18 000 mg/m^3 (10 000 ppm(v)) (EPA 1991). Indoor carbon dioxide concentrations will not reach these levels unless the ventilation rate is extremely low, about 1 L/s (2 cfm) per person for 9000 mg/m^3 (5000 ppm(v)) and less than about 0.2 L/s (0.4 cfm) per person for 54 000 mg/m^3 (30 000 ppm(v)).

The association between carbon dioxide concentrations and occupant perceptions of the indoor environment in terms of comfort and irritation is complex because it mixes several different issues, including the comfort impacts of the carbon dioxide itself, associations between carbon dioxide levels and the concentrations of other occupant-generated contaminants, and the relationship between carbon dioxide and ventilation. Some indoor air quality investigators associate indoor carbon dioxide concentrations from 1100 mg/m^3 (600 ppm(v)) to 1800 mg/m^3 (1000 ppm(v)) or higher with perceptions of stuffiness and other indicators of discomfort and irritation (Bright et al. 1992; Rajhans 1983; Bell and Khati 1983). However, these associations are often based on anecdotal observations of the investigator or on informal occupant surveys. Seppanen et al. (1999) reviewed twenty-one studies, involving more than thirty thousand subjects, of ventilation rates, indoor CO_2 levels and sick building syndrome (SBS) symptoms and found that higher CO_2 levels were associated with increased symptoms in about half of the studies. The authors do note that there were significant variations among the studies in the CO_2 metric employed and a number of measurement issues. Also, they note that it is unlikely that the symptoms were associated with exposure to CO_2, but rather to other contaminants. Apte et al. (2000) examined data from forty-one U.S. office buildings from the EPA BASE study in which the measurement protocol was standardized and probability sampling was used to select the buildings. Significant associations were seen between some SBS symptoms and CO_2 levels, though the authors acknowledge that CO_2 is likely a surrogate for other occupant-generated pollutants and for the ventilation rate per occupant. In other words, CO_2 levels increase with increased occupancy and decreased ventilation rate, and it may be these latter two factors that are actually causing the symptoms. In addition, the observed associations between carbon dioxide and occupant comfort may be due to other factors, such as thermal comfort or the concentrations of other contaminants in the space. However, as discussed below, there is a demonstrated correlation between indoor carbon dioxide concentrations and the level of acceptability of the space in terms of human body odor.

The relationship between carbon dioxide concentrations and the concentrations of other indoor contaminants depends on the characteristics of the sources of these other contaminants. As discussed earlier, the rate at which carbon dioxide is generated in a space depends on the number of people in the space, their size and their level of physical activity. If other contaminants are generated at a rate that also depends on these factors, then carbon dioxide may be a good indicator of their concentrations. However, only some indoor contaminants are generated at a rate that depends on occupancy, and many are not a function of occupancy at all. For example, emissions from building materials and furnishings, the intake of outdoor contaminants by the ventilation system, and contaminants associated with some occupant activities do not depend on the number of occupants in a space. Regardless of the indoor carbon dioxide level, the concentration of contaminants emitted by occupant-independent sources may be high, low or in between and the carbon dioxide concentration will not provide any information on their concentration. This fact is one limitation on the use of carbon dioxide based demand controlled ventilation.

Carbon Dioxide Concentrations and Body Odor Acceptability

At the same time people are generating CO_2, they are also producing odor-causing bioeffluents. Similar to carbon dioxide generation, the rate of bioeffluent generation depends on the level of physical activity. Bioeffluent generation also depends on diet and on personal hygiene. Because both carbon dioxide and bioeffluent generation rates depend on physical activity, the concentrations of carbon dioxide and the odor intensity from human bioeffluents in a space exhibit a similar dependence on the number of occupants and the outdoor air ventilation rate.

Experimental studies have been conducted in chambers and in occupied spaces, in which people evaluated the acceptability of the air in terms of body odor (Berg-Munch et al. 1986; Cain et al.

1983; Fanger and Berg-Munch 1983; Fanger 1988; Iwashita et al. 1990; Rasmussen et al. 1985). These experiments studied the relationship between outdoor air ventilation rates and odor acceptability, and are a major consideration in developing the ventilation rate recommendations in ventilation standards. Some of the experiments also studied the relationship between the acceptability of the air in the space in terms of odor and carbon dioxide concentrations.

These studies have concluded that about 7 L/s (15 cfm) of outdoor air ventilation per person will control human body odor such that roughly 80 % of unadapted persons (visitors) will find the odor at an acceptable level. The same level of odor acceptability was found to occur at carbon dioxide concentrations that are about 1250 mg/m^3 (700 ppm(v)) above the outdoor concentration, which at a typical outdoor level of 630 mg/m^3 (350 ppm(v)) yields an indoor carbon dioxide concentration of 1880 mg/m^3 (1050 ppm(v)). Based on these considerations, 1800 mg/m^3 (1000 ppm(v)) carbon dioxide is a commonly discussed guideline value (ASHRAE 1989). The differential between indoor and outdoor levels of 1250 mg/m^3 (700 ppm(v)) is a measure of acceptability with respect to body odor, irrespective of the outdoor carbon dioxide concentration. Figure 2 shows the percent of unadapted persons (visitors) who are dissatisfied with the level of body odor in a space as a function of the carbon dioxide concentration above outdoors (CEC 1992). People adapt quickly to bioeffluents. For adapted persons (occupants), the ventilation rate per person to provide the same acceptance is approximately one third of the value for unadapted persons (visitors) and the corresponding carbon dioxide concentrations above outdoors are three times higher (Berg-Munch et al. 1986; Cain et al. 1983).

The relationship between percent dissatisfied and carbon dioxide concentrations for visitors shown in Figure 2 was seen experimentally (Berg-Munch et al. 1986; Fanger and Berg-Munch 1983; Rasmussen et al. 1985), and the correlation was not strongly dependent on the level of physical activity. In addition, the relationship did not require that the indoor carbon dioxide concentration be at equilibrium. The relationship described in Figure 2 can also be derived based on the experimentally-determined relationship between percent dissatisfied and outdoor air ventilation rates in L/s (cfm) and the relationship between outdoor air ventilation rates and equilibrium carbon dioxide concentrations that is described later in this paper.

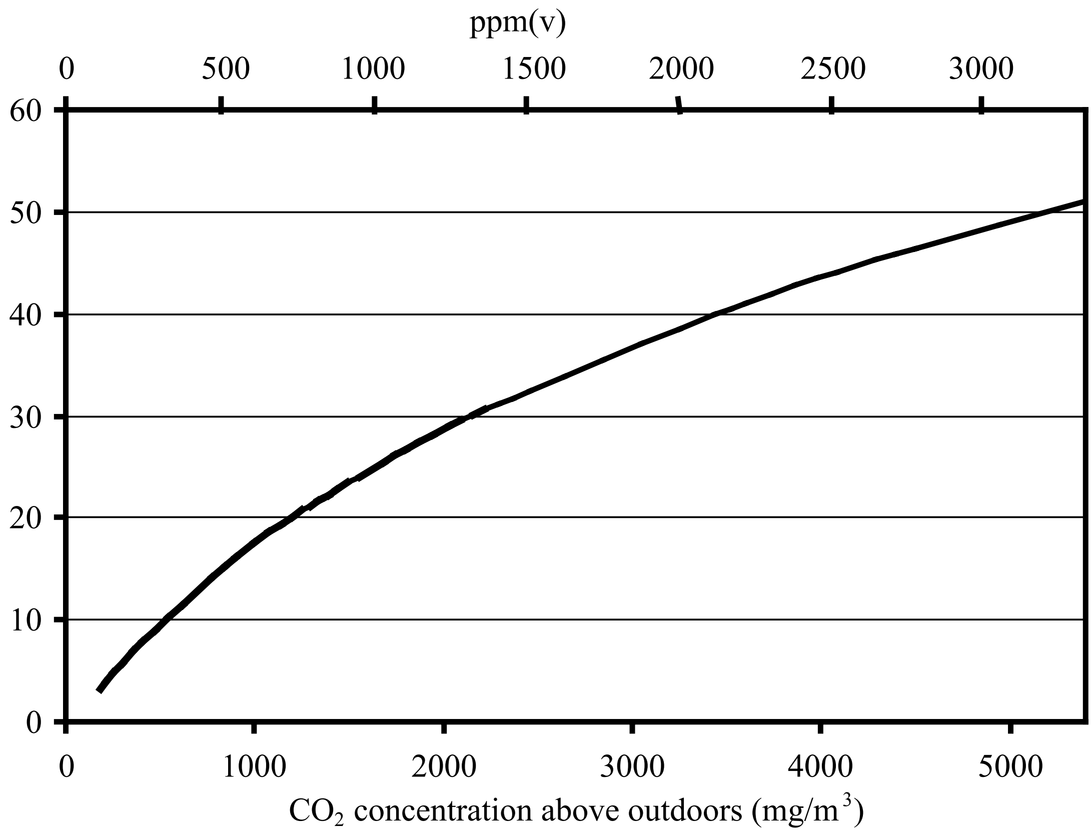


Figure 2 Percent of Visitors Dissatisfied with Bioeffluents Odor as a Function of CO_2 Concentration (CEC 1992)

While carbon dioxide concentrations can be an appropriate means of characterizing the acceptability of a space in terms of body odor, as stated earlier, they do not provide information on the control of contaminants from other pollutant sources such as building materials, furnishings and occupant activities. And while maintaining carbon dioxide concentrations within 1250 mg/m^3 (700 ppm(v)) of outdoors should provide acceptable perceived air quality in terms of human body odor, it does not necessarily imply adequate control of these other pollutant sources.

Some have viewed this relationship of CO_2 with bioeffluents perception as a problem with CO_2 DCV, reasoning one can only control the level of odor from bioeffluents with this approach. This conclusion is erroneous, since CO_2 can also serve as an indicator of ventilation per person independent of the relationship seen in Figure 2.

Carbon Dioxide and Ventilation Rates

The relationship between carbon dioxide and outdoor air ventilation rates is fairly well understood (Persily and Dols 1990; Persily 1997). All else being equal, if the ventilation rate in an occupied space decreases then the carbon dioxide concentration will increase. However, making quantitative estimates of building ventilation rates based on measured CO_2 concentrations requires the use of a specific technique that is appropriate to the conditions that exist in the building, and is not always as simple as has sometimes been suggested (Persily 1997). Fortunately, the use of CO_2 DCV does not rely on making estimates of building ventilation rates based on CO_2 concentrations. In the context of this report, the relevant issues of the relationship between indoor CO_2 levels and ventilation include steady-state CO_2 concentrations at a constant air change rate and the time it takes to achieve steady-state conditions.

Steady-state CO_2 concentrations can be determined for a given ventilation rate based on a single-zone mass balance analysis. Assuming that the CO_2 concentration in the building or space of interest can be characterized by a single value C, the mass balance of CO_2 can be expressed as follows:

$$V\frac{dC}{dt} = G + QC_{out} - QC \tag{4}$$

where

V = building or space volume (mass) in m^3 (mg)
C = indoor CO_2 concentration in mg/m^3 (ppm(v))
C_{out} = outdoor CO_2 concentration in mg/m^3 (ppm(v))
t = time in s
G = indoor CO_2 generation rate in mg/s (m^3/s)
Q = building or space ventilation rate in mg/s (m^3/s)

For a constant generation rate (occupancy level) and constant ventilation rate and outdoor concentration, the indoor concentration will eventually attain a steady state or equilibrium concentration C_{ss} given by the following expression:

$$C_{ss} = C_{out} + G/Q \tag{5}$$

If the generation rate G and the ventilation rate Q are expressed in L/s (as in Figure 1), and the concentrations are in mg/m^3, then Equation (5) takes the form:

$$C_{ss} = C_{out} + \frac{1.8\times10^6 G}{Q} \tag{6}$$

If instead G and Q are expressed in cfm and the concentrations are in ppm(v), then Equation (5) takes the form:

$$C_{ss} = C_{out} + \frac{10^6\times G}{Q} \tag{7}$$

Therefore, as the ventilation rate increases, the steady-state CO_2 concentration decreases.

Assuming the building or space begins the day at the outdoor CO_2 concentration and is then occupied, the indoor concentration will start to rise at a rate that depends on the building ventilation rate Q divided by the building volume V. This quantity, Q/V, is sometimes referred to as the outdoor air change rate of the building, while its inverse V/Q is sometimes referred to as the time constant of the system. During this build-up, the indoor CO_2 concentration is governed by the transient solution to Equation (4):

$$C(t) = C_{out} + \frac{G}{Q}\left(1 - e^{\frac{-Qt}{V}}\right) \tag{8}$$

Note that as t approaches infinity, the concentration C approaches the steady-value given in Equation (5) as expected. It is also important to note that the time required to reach steady-state depends on the value of Q/V, with higher values (higher air change rates) corresponding to less time required to approach steady-state. Figure 3 is a plot of the build-up in indoor CO_2 concentration, calculated from Equation (8), for different air change rates. Figure 3 is based on a generation rate of 0.0052 L/s (0.011 cfm) per person and an outdoor concentration of 630 mg/m^3 (350 ppm(v))

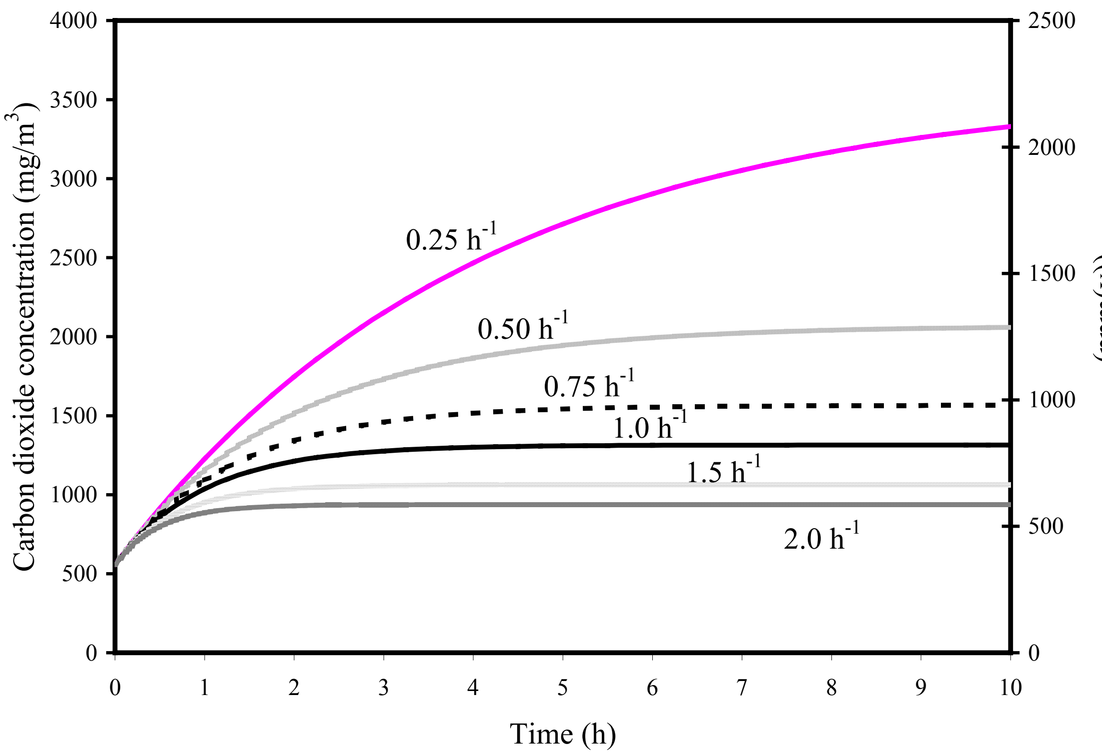


Figure 3 Calculated Carbon Dioxide Build-up as a Function of Air Change Rate

Figure 3 depicts the time required for indoor CO_2 to reach steady-state concentration. Somehave identified this delay as a problem with the application of CO_2 demand controlled ventilation. However, since DCV need not be based on the relationship between steady-state CO_2 levels and ventilation rates, this buildup time is not a problem.

Carbon Dioxide Control versus Ventilation Control

Some discussions, and criticisms, of CO_2 demand controlled ventilation are based on the inadequacy of CO_2 as an overall indicator of indoor air quality. This limitation has been noted above and is based on many important contaminants not being generated on a per-person basis, for example, building materials. However, the application of CO_2 DCV is better understood based on its use as an indicator of ventilation rate per person. Specifically, the control approach is more appropriately based on the desire to maintain a specific outdoor airflow rate per person based on a building code or ventilation standard. If the ventilation rate per person is lower than desired, the CO_2 level will build up above its setpoint, or at a rate that is recognized as high, and the control system will need to respond by increasing the ventilation rate. If the ventilation rate is higher than required based on the design value of outdoor air per person, the CO_2 level will be lower than the target and the control system can respond by decreasing the ventilation rate. That decrease is the mechanism by which CO_2 DCV realizes energy savings. However, the control system need not wait for the CO_2 level to reach its steady-state value to make this decision. Control algorithms can be employed that anticipate where the CO_2 level is headed and make adjustments to the ventilation rate well in advance of steady-state conditions. Furthermore, since other indoor contaminants buildup over time rather than instantaneously, CO_2 control can be used to take advantage of this transient effect by lagging the start of ventilation for a period of time, thereby realizing additional energy savings. Therefore, it is important to realize that CO_2 DCV uses indoor carbon dioxide to control ventilation and that the objective is not simply to control the indoor CO_2 level.

3. LITERATURE REVIEW

In the last fifteen years, interest in CO_2-based DCV has led to a large body of literature published in journals, conference proceedings, and other forums. An extensive literature review (Raatschen 1990) covering all aspects of demand controlled ventilation, including non-CO_2-based systems, was published at the conclusion of Annex 18, an International Energy Agency effort to develop guidelines for DCV systems. This supercedes a more limited review published during Annex 18 by Mansson (1989). The increasing interest in DCV in the U.S. is evidenced by recent articles published in several trade journals (Wright 1997; Di Giacomo 1999; Schell and Int-Hout 2001; and Schell 2001). The objective of this section, which is an update of an earlier report (Emmerich and Persily 1997), is to summarize the literature on CO_2-based DCV as applied to non-residential buildings.

Literature reports on CO_2-based DCV are categorized in this paper as follows: Case Studies-Field Tests; Case Studies-Simulations; Sensor Performance and Location; and Application. The first two categories include studies of the performance of CO_2-based DCV systems in real buildings and using computer models. The various case studies that have been conducted focus on issues including ventilation rates, energy consumption, economic impacts and the concentrations of other indoor pollutants, though few studies address all of these issues. The third category includes reports that address the performance of CO_2 sensors and where they should be located in a space. The fourth category discusses the application of CO_2-based DCV, from very general descriptions to detailed discussions of control algorithms.

3.1 Case Studies-Field Tests

There have been many demonstration projects in which CO_2-based DCV systems were installed in buildings and some aspects of performance were monitored. These studies vary in many respects, including the detail with which the DCV systems are described. Some reports contain detailed descriptions of the DCV control algorithms, while others do not even report the setpoint. The studies also vary in the impacts that were monitored, which have included fan operation, damper position, indoor CO_2 concentrations, ventilation rates, energy consumption, the concentrations of other pollutants, and occupant perceptions of the indoor environment. Finally, the studies have taken place in a variety of building types including offices, schools, auditoria and retail stores.

The application of CO_2-based DCV is often discussed with reference to office buildings, and occasionally to conference rooms within office buildings. One of the earliest studies of CO_2 control in an office building took place in Helsinki (Sodergren 1982). The outdoor air control algorithm is not described, but the CO_2 setpoint was 1260 mg/m^3 (700 ppm(v)). The CO_2 control system was compared to constant outdoor air and timer-based control, and 24-h plots of CO_2 concentration are presented for each system. Measured concentrations of other pollutants and interviews with occupants did not indicate any IAQ problems.

Davidge (1991) presents a demonstration project in a 30,000 m^2 (320,000 ft^2) Canadian office building. In this building, the system never reduced the ventilation rate because the outdoor temperatures in the winter were never low enough to go off free-cooling. During the summer, damper leakage was more than enough to control CO_2. Davidge also studied a boardroom, where supplemental ventilation was controlled alternatively by a light switch, a motion sensor and a CO_2 controller. In the case of the CO_2 controller, the fan came on at 1440 mg/m^3 (800 ppm(v)) and shut off at 1080 mg/m^3 (600 ppm(v)). An occupant questionnaire was administered, and it was found that the occupants could not distinguish whether or not the fan was on in terms of air quality. However, they rated the CO_2 system very highly, presumably in terms of indoor air quality though the report does not specify the survey results in any detail.

A fairly comprehensive study of CO_2 control took place on two floors of an office building in Montreal (Donnini et al. 1991, Haghighat and Donnini 1992). One floor was equipped with a CO_2 DCV system, while the other floor served as a control. The CO_2 control algorithm was as follows: the damper closed at concentrations below 1080 mg/m^3 (600 ppm(v)); as CO_2 increased above 600 ppm(v) the dampers opened with the maximum opening at 1800 mg/m^3 (1000 ppm(v)). The study lasted one year, during which indoor concentrations of CO_2, formaldehyde, volatile organic compounds and particles, ventilation system performance, thermal comfort, and occupant perception were measured once a month. Energy demand was monitored for the whole year. The outdoor air dampers were closed most of the year, because there were rarely enough people to raise the indoor CO_2 concentration. The indoor air quality measurements revealed no significant contaminant concentration differences between the CO_2 and the control floor. Thermal comfort was generally adequate on both floors. Annual energy savings of 12 % were measured for the floor with DCV. Occupants of the DCV floor complained significantly more about the indoor environment than occupants of the control floor.

Fleury (1992) reported on the performance of a CO_2 controlled ventilation system in a conference room. In this system, the fan motor speed was adjusted according to the CO_2 concentration, but no information was provided on the specific control algorithm or setpoints. The measured CO_2 concentrations in the space were between 630 mg/m^3 (350 ppm(v)) to 1530 mg/m^3 (850 ppm(v)), with one peak of 1980 mg/m^3 (1100 ppm(v)). Based on occupant questionnaires, the air quality was rated from good to excellent. Another study was undertaken in a conference room set up to test DCV sensors, including CO_2, volatile organic compounds and humidity (Ruud et al. 1991). The CO_2 setpoints were not reported, but the indoor concentration never exceeded 1620 mg/m^3 (900 ppm(v)). Another demonstration in a conference room is reported by Huze et al. (1994). The ventilation rate was varied proportionally to the CO_2 concentration within a 900 mg/m^3 (500 ppm(v)) band centered around 2160 mg/m^3 (1200 ppm(v)). Limited results presented include a sample of the CO_2 level and control signal for one day.

One of the most frequently cited demonstration projects took place in a small bank in Pasco, Washington (Gabel et al. 1986). This study involved the measurement of energy consumption, contaminant levels including nitrogen dioxide, formaldehyde, carbon monoxide and particulates, and occupant response based on a questionnaire. The study design included monitoring over the winter, spring and summer seasons, with one week of normal operation followed by one week of CO_2 control. The system's economizer cycle operated normally throughout the test periods. They found that with the CO_2 control system setpoint at 1800 mg/m^3 (100 ppm(v)) to 2160 mg/m^3 (1200 ppm(v)), air leakage through the closed damper provided sufficient fresh air for typical occupancy, which was only 10 % to 15 % of design. That is, the indoor CO_2 level never rose to the control setpoints. All measured contaminants were maintained below indoor standards. Based on a curve fit of the measured energy consumption to outdoor temperature for the two modes of outdoor air control, average energy savings of 7.8 % for heating and cooling in six climates typical of Oregon and Washington were calculated. Based on the questionnaires, the occupants could not detect differences between background CO_2 levels of 540 mg/m^3 (300 ppm(v)) and 1800 mg/m^3 (100 ppm(v)). The occupants reported feeling warmer during DCV control, although the measured indoor temperatures were no different.

Another frequently-cited study took place in a Minnesota high school (Janssen et al. 1982). The ventilation system used CO_2 and temperature to control outdoor air, and had separate dampers for temperature and CO_2 control. Indoor contaminants, energy and subjective response of occupants were monitored. The measured energy savings were about 20 %. The occupant questionnaire showed that the subjects felt warmer with increased CO_2 concentrations. Another study by the same group of researchers took place in a portion of a high school, which was retrofitted with a CO_2-controlled system (Woods et al. 1982). During the early months of 1980, the system operated under alternate periods with conventional temperature control and with CO_2 control. System performance was monitored, and the subjective responses of occupants were obtained. The system contained a set of outdoor air dampers that were controlled based on the CO_2 concentration. These dampers modulated between fully closed and fully open damper positions, with the low setpoint at 5400 mg/m^3 (3000 ppm(v)) and the high setpoint at 9000 mg/m^3 (5000 ppm(v)). The results indicated the potential for significant energy savings. Occupants felt warmer when CO_2 control operated despite the fact that there was no measurable temperature difference with and without CO_2 control.

A study of two Finnish public buildings, one of which had CO_2 controlled ventilation, included measurements of radon, particles and CO_2 (Kulmala et al. 1984). No description of the CO_2 control algorithm was reported. Daily energy savings were estimated at 13 % to 20 %.

In several of the studies cited so far, the indoor CO_2 concentration was often not high enough for the CO_2 control system to operate. This may be due in part to the relatively low occupant density in office buildings. The application of CO_2-based DCV is usually viewed as better suited to spaces where occupancy is more variable and where the peaks are associated with fairly high occupancy. Auditoria are good examples of such spaces, and there have been several case studies in these types of spaces. One such study took place in an auditorium with CO_2 and timer control of ventilation at the Swiss Federal Institute of Technology in Zurich (Fehlmann et al. 1993). The measurements included system run time, energy use, climatic parameters and CO_2 concentrations under winter and summer conditions. In addition, an occupant questionnaire was administered. The ventilation system had two stages of airflow capacity, with the first stage coming on at 1080 mg/m^3 (600 ppm(v)) and the second stage at 2340 mg/m^3 (1300 ppm(v)). The second stage would turn off at 1980 mg/m^3 (1100 ppm(v)), and the first stage at 1080 mg/m^3 (600 ppm(v)). With ventilation controlled by CO_2, run time was 67 % of the run time with timer control in summer and 75 % in winter. Energy consumption with CO_2 control was 80 % less in summer and 30 % less in winter. Questionnaire results indicated a higher perception of odors with CO_2 control, especially in the summer. It was noted that the occupancy was very low compared to design, only about 10 % to 20 %.

Zamboni et al. (1991) reported on field measurements in auditoria in Norway and Switzerland. In the Norwegian building, the CO_2 setpoint was 1800 mg/m^3 (1000 ppm(v)), and the reported results include indoor temperature, CO_2 concentration and age of air. In the Swiss building, there was a two-stage controller with the first setpoint at 1350 mg/m^3 (750 ppm(v)) and the second at 2340 mg/m^3 (1300 ppm(v)). The researchers monitored energy consumption and indoor climate, and administered occupant questionnaires. Heating energy was reduced by 15 % during one week of testing in the winter and by 75 % in the summer. With CO_2 control, there was less draft but more odor in summer.

Several demonstration projects have been conducted in so-called public spaces, including retail stores and recreational facilities, where occupancy is expected to be more variable and less predictable. Potter and Booth (1994) report on the performance of CO_2-based DCV systems in eight public buildings. The authors note that the results point to some potential problems with CO_2 control, but many of the results are presented simply in the form of plots of indoor CO_2 concentration versus time. In an office building and a swimming pool facility, the indoor concentration never reached the CO_2 setpoint. Building setpoints were variable and included 2250 mg/m^3 (1250 ppm(v)), 3960 mg/m^3 (2200 ppm(v)) and 4500 mg/m^3 (2500 ppm(v)). Based on the results, the authors identify candidate building types for CO_2 control as cinemas, theatres, bingo and snooker establishments, educational lecture theatres, teaching labs, meeting rooms, and retail premises. They considered the issues of maintenance and reliability, noting that no controllers in the buildings were marked for calibration due date or the date of last service.

Another study of two public spaces took place in a social club and a cinema in England (Anon 1986). The control algorithm was not described, but the CO_2 setpoints were usually between 1260 mg/m^3 (700 ppm(v)) and 1800 mg/m^3 (1000 ppm(v)). The measured fuel savings were 17 % in the club and 11 % at the cinema. Warren (1982) reports on tests of energy savings with CO_2 control in a theater and a retail store. Energy and cost savings estimates are based on short term tests in the building, and the dependence of the savings on ventilation system design parameters is discussed. The systems in the two buildings are not described in detail.

Chan et al. (1999) address the case of a lecture theater in Hong Kong where radon is known to be of concern. They propose a DCV system controlled by both CO_2 and radon measurements to achieve acceptable IAQ while saving energy. Few details are presented.

Finally, Strindehag et al. (1990) and Strindehag and Norell (1991) reported on a number of examples of how outdoor air intake can be controlled by CO_2 in a conference room, an auditorium, three offices and a school. The report contains descriptions of the buildings and the CO_2 sensors, and notes that the CO_2 setpoint was 1080 mg/m^3 (600 ppm(v)). However, the control algorithms are not described, and no specific performance indicators are discussed. Satisfactory reliability of the system in the auditorium was reported after three years of operation.

The studies cited here show that CO_2 control has been demonstrated in a wide variety of building types including offices, schools, and public. It is apparent in examining these studies that the CO_2 control algorithm is often not described in sufficient detail to understand the system; in fact, some of the studies did not even report CO_2 setpoints. In several of the demonstration projects, the building occupancy was insufficient to raise the indoor CO_2 concentration enough to activate the CO_2 control system. Several of the studies used occupant questionnaires to evaluate performance, with inconsistent results. In some cases, the occupants perceived the indoor environment with CO_2 control positively. In other cases, there were more complaints, specifically with regards to odor during CO_2 control. Several studies noted a feeling of increased warmth with elevated CO_2 concentration despite the fact that the measured indoor temperatures were no higher. When considering these reports of occupant response, it must be kept in mind that the studies employed different questionnaires.

3.2 Case Studies-Simulations

As discussed above for field tests, the reported simulation case studies vary widely in both the description of important parameters and discussion of results. Most studies have focused on the potential energy savings of the CO_2-based DCV systems, with CO_2 concentrations reported as a measure of IAQ performance. A few studies have calculated concentrations of other pollutants. As with the field tests, the majority of the studies have involved office buildings, with others examining schools, retail buildings, restaurants and auditoria. Another important issue in simulations is the treatment of infiltration and interzone airflows, with most studies using assumed rates and a few studies employing a multizone airflow model.

Recently, Brandemuehl and Braun (1999) investigated the energy impact of various combinations of six economizer and DCV strategies (no economizer, dry bulb economizer, and enthalpy economizer – each with and without DCV) for four types of buildings (office, large retail store, school, and sit-down restaurant) in twenty U.S. climates (including Los Angeles and Sacramento). Additional modeling assumptions included a CO_2 setpoint of 1260 mg/m^3 (700 ppm(v)) above ambient, thermostat setup or setback at night, HVAC fan shutdown during unoccupied hours, single-zone buildings with no infiltration, ventilation effectiveness of 0.85, and minimum ventilation flows for non-DCV cases of 9.4 L/s (20 cfm) per person, 4.7 L/s (10 cfm) per person, 7.1 L/s (15 cfm) per person, and 9.4 L/s (20 cfm) per person for the office, retail, school, and restaurant cases, respectively. The DCV system resulted in significant reductions in heating energy use for all buildings and climates. Heating energy use reductions ranged from 40 % for the office to 100 % for the retail building (i.e., the solar and internal loads supplied all necessary heat) in Sacramento and from 75 % for the office to 100 % for the retail building in Los Angeles. The DCV system with enthalpy economizer required the least cooling energy use for all building types and climates. However, in some cases, much of the cooling energy reduction was due to the economizer, and use of DCV without an economizer can actually increase cooling energy use for dry climates. Cooling energy reductions ranged from about 10 % to 20 % for all buildings in Sacramento and Los Angeles. The authors also note that the savings associated with DCV are very dependent on the occupancy schedule and its relationship to the design occupancy used to set the fixed minimum ventilation rate of the base case. Also, for some types of buildings, additional ventilation may be required to maintain other contaminants at acceptable levels.

In an early report of a simulation study for an office, Knoespel et al. (1991) investigated the application of a CO_2-based DCV system to a two-zone office space with both constant air volume (CAV) and variable air volume (VAV) HVAC systems. A multiple zone pollutant transport model was used and a ventilation airflow controller model was developed as modules for a transient thermal system simulation program (Klein 1994). Other existing modules of the program were used to calculate building energy consumption. Infiltration to the main zone was assumed constant at 0.2 h^{-1} and an interzone flow of 12 L/s (24 cfm) from the main office to the conference room was included when the HVAC system was on. Knoespel compared the performance of six ventilation strategies including constant outdoor airflow at the ASHRAE Standard 62-1989 prescribed flow of 10 L/s (20 cfm) per person, constant outdoor airflow at a "typical" rate of 0.7 h^{-1}, minimum outdoor airflow at the typical rate with a temperature-based economizer, DCV with a step-flow control algorithm, DCV with step-flow control and a temperature-based economizer, and DCV with on-off control. In the step-flow control algorithm, the fraction of outdoor air in the circulation flow was changed in 20 % steps depending on whether the measured CO_2 concentration in either zone was above or below the specified limit. On-off control employed an algorithm in which outdoor airflow is set at 100 % if the high CO_2 setpoint is exceeded and at 0 % if the CO_2 concentration drops below the low setpoint. The setpoints used were 1440 mg/m^3 (800 ppm(v)) and 1800 mg/m^3 (1000 ppm(v)). Simulations were performed for Miami, FL and Madison, WI. In Madison, the DCV strategies provided acceptable control of CO_2 levels with coil energy savings from 9 % to 28 % for CAV systems and from 43 % to 46 % for VAV systems compared to the Standard 62-1989 prescribed rate strategy. The savings for Miami were of similar absolute magnitude but smaller percentages. These results did not include fan energy use. Compared to the economizer and constant outdoor airflow strategies at typical rates, the DCV strategies resulted in similar energy use with better control of CO_2 concentrations for both CAV and VAV systems.

Emmerich et al. (1994) applied the model developed by Knoespel et al. (1991) to examine the performance of DCV systems under less favorable conditions and to study the impact on non-occupant generated pollutants. Emmerich used the same building, Madison location, and the HVAC systems described above but varied the simulated conditions to include a pollutant removal effectiveness as low as 0.5 and an occupant density up to 50 % greater than design. For all cases examined, the DCV system reduced the annual cooling and heating loads from 4 % to 41 % while maintaining acceptable CO_2 concentrations. In addition to requiring more energy use, the constant outdoor airflow strategy resulted in CO_2 levels above 1080 mg/m^3 (600 ppm(v)) for more than half of occupied hours for cases with poor pollutant removal effectiveness. Emmerich also examined the impact of DCV on non-occupant generated pollutants by modeling a constant source of a non-reactive pollutant located in the main office zone. Four ventilation strategies were compared including constant outdoor air at a prescribed rate based on ASHRAE Standard 62-1989, DCV with step control and setpoints of 1440 mg/m^3 (800 ppm(v)) and 1800 mg/m^3 (1000 ppm(v)), DCV with a constant minimum outdoor airflow rate of 2.5 L/s (5 cfm) per person calculated using the multiple space method of ASHRAE Standard 62-1989, and DCV with scheduled purges of 100 % outdoor air from 7:30 a.m. to 8:30 a.m. and 12:30 p.m. to 1:00 p.m. The non-occupant generated pollutant source strength was specified such that the system with constant outdoor airflow rate just met a short-term limit of 2 ppm(v) and an 8-h average limit of 1 ppm(v). (These concentrations cannot be converted to SI units, since this generic contaminant is not associated with any specific molecular weight.) Emmerich found that both the straight DCV and the DCV with minimum outdoor airflow rate failed to meet the pollutant concentration limits for both the CAV and VAV systems, but the DCV with scheduled purge strategy successfully limited the pollutant concentrations. The purge strategy increased building heating and cooling loads over the straight DCV strategy but still reduced the loads by 17 % (CAV) and 25 % (VAV) compared to the constant outdoor airflow case. The success of the purge strategy was attributed partially to the ability to schedule the purges when most needed.

In another study considering the effects of poor ventilation air mixing, Haghighat et al. (1993) simulated the performance of a CO_2-based DCV system in a large office building in Montreal. The baseline ventilation system had a flow rate of 10 L/s (20 cfm) per person, and a mixing parameter of 0.7 was used in the model. The DCV system used a minimum ventilation rate of 2.5 L/s (5 cfm) per person, and the ventilation rate was adjusted each hour to maintain a CO_2 concentration of 1440 mg/m^3 (800 ppm(v)). Infiltration was 0.4 h^{-1} with the HVAC system off and 0.04 h^{-1} with it on. Four cases of occupant density were examined. The DCV system saved from 7 % to 15 % in energy use, 2 % to 6 % in energy cost, and 7 % to 17 % in peak demand compared to a fixed ventilation rate strategy. In a follow-up study using the same office model with different infiltration, operating hours and other assumptions, Zmeureanu and Haghighat (1995) found energy consumption for the DCV system ranging from a 5 % decrease to a 2 % increase. However, because of peak demand reductions, annual energy cost savings ranging from 3 % to 26 % were found.

Sorensen (1996) also describes simulations performed for a two-zone office with a conference room. A unique aspect of this study is its focus on examining the short term dynamics of the system by simulating a ten hour period with one second time steps and detailed modeling of the HVAC system. A VAV system with dual temperature and CO_2 control and CAV system without CO_2 control are simulated. Because a detailed VAV system model is used, the control algorithm is more complex than in most studies reviewed and involves both dampers and fans. When the CO_2 concentration is above an upper limit of 1620 mg/m^3 (900 ppm(v)), the damper actuator position increases by 1 %. If the concentration remains above the upper limit, the position continues to increase until it is fully open or until it drops below the limit. After the damper is fully open, a concentration above the upper limit will increase the fan speed by 5 % until the fan reaches maximum speed or the concentration falls below the limit. The algorithm also uses a lower limit of 1260 mg/m^3 (700 ppm(v)) to decrease fan speed and damper position. Detailed results are not presented, but transient CO_2 concentrations and temperatures are presented and it is stated that the VAV system used 31 % less energy than the CAV system for a cold ambient condition.

Another recent study of office applications (Carpenter 1996 and Enermodal 1995) examined both the energy and IAQ impacts of CO_2-based DCV in a mid-sized commercial building complying with ASHRAE Standard 90.1 in four climate zones (Chicago, Nashville, Phoenix, and Miami). Simulations were performed using a combination of an energy analysis program (Enermodal 1990) and the multizone pollutant transport program CONTAM87 (Axley 1988). Three HVAC systems (single-zone, multizone, and VAV) and 5 ventilation control strategies (fixed ventilation rate, DCV with building return air controlled to 1800 mg/m^3 (1000 ppm(v))and 1440 mg/m^3 (800 ppm(v)), DCV with floor return controlled to 1000 ppm(v), and DCV with each zone controlled to 1800 mg/m^3 (1000 ppm(v)) were analyzed. The DCV control algorithm was not described in detail. For single-zone systems, the DCV strategy reduced heating energy by about 30 % for a setpoint of 1800 mg/m^3 (1000 ppm(v)) and by 20 % for a setpoint of 1440 mg/m^3 (800 ppm(v)). The DCV system with a setpoint of 800 ppm(v) also reduced average CO_2 concentrations by 90 mg/m^3 (50 ppm(v)) to 160 mg/m^3 (90 ppm(v)) compared to the fixed ventilation rate strategy. The DCV strategies had little effect on cooling energy, because the DCV system tended to reduce ventilation during the cooler morning and evening hours and increase ventilation during the warmer middle of the day. For VAV systems, the energy savings were similar to those with single-zone systems. For multizone systems, the reduction in heating energy was similar in absolute terms but was smaller in percent (5 % to 12 %) because of a larger total heating load. DCV with a setpoint of 1800 mg/m^3 (1000 ppm(v)) resulted in average CO_2 concentrations 130 mg/m^3 (70 ppm(v)) to 270 mg/m^3 (150 ppm(v)) higher than the fixed ventilation strategy, while a setpoint of 1440 mg/m^3 (800 ppm(v)) kept concentrations lower than the fixed strategy and the maximum below 1800 mg/m^3 (1000 ppm(v)) in all zones. Providing additional sensors in return duct of each floor had little impact on energy use and IAQ. Installing sensors in each zone ensured that the concentration in each zone stayed below 1800 mg/m^3 (1000 ppm(v)) but at a slightly higher energy use. The performance of DCV with sensors set at 1800 mg/m^3 (1000 ppm(v)) in each zone was similar to central control with a setpoint of 1440 mg/m^3 (800 ppm(v)). Formaldehyde concentrations were also simulated to evaluate the impact of DCV strategies on pollution from a non-occupant source. None of the DCV strategies controlled the formaldehyde concentrations as well as the fixed ventilation strategy. It was suggested that a morning purge should be included in a DCV strategy when non-occupant generated pollutants are a concern, but this option was not simulated. Different DCV control algorithms including on-off, linear proportional, proportional-integral-derivative (PID), and the Vaculik method (discussed later in this paper) were discussed but not simulated.

Wang and Jin (1998) also simulate the performance of CO_2-based DCV for an office with a focus on describing and evaluating a control algorithm that can adjust ventilation rates based on estimated occupancy. Three different occupancy estimation algorithms (steady state, approximate dynamic detection, and exact dynamic detection) were compared for a single well-mixed zone with three different occupant densities and patterns. Both dynamic detection methods detected occupancy with high accuracy and the change of occupancy with a fast response time. Later, Wang and Jin (1999) experimentally verified the capability of the algorithms.

Wang and Jin then performed simulations to compare the IAQ and energy performance of four ventilation strategies (DCV using approximate dynamic detection algorithm, DCV with a CO_2 upper limit of 1800 mg/m^3 (1000 ppm(v)), DCV with CO_2 upper limit of 1800 mg/m^3 (1000 ppm(v)) upper limit and 1440 mg/m^3 (800 ppm(v)) lower limit, and constant outdoor air) for an eight-zone open-plan office with two different occupant densities. The office had a combination of CAV and VAV ventilation systems. Simulations were performed for single days of summer and spring Hong Kong weather. The study found the two DCV systems based on direct CO_2 measurement were able to control CO_2 levels as well as the occupancy detection method but could not maintain constant ventilation rates per occupant. Unfortunately, the CO_2 concentrations results indicate a potential major flaw in the model assumptions. At the end of the day, the ventilation system is turned off and there is no infiltration overnight resulting in initial CO_2 concentrations of 1440 mg/m^3 (800 ppm(v)) to 1800 mg/m^3 (1000 ppm(v)). This assumption masks likely significant differences in system performance during the morning hours as CO_2 concentration increases from background levels. Also, the authors conclude that this would result in inadequate indoor air quality but provide no justification. Compared to the constant ventilation strategy, all three DCV strategies were found to reduce coil loads about 8 % for spring weather and from 12 % to 18 % for summer weather.

Meckler (1994) also simulated the application of CO_2-based DCV in an office building. The energy performance of an idealized DCV system with the ventilation rate varied to maintain 1440 mg/m^3 (800 ppm(v)) and 1660 mg/m^3 (920 ppm(v)) (i.e., no control algorithm modeled) was compared to a baseline system with a constant ventilation rate of 10 L/s (20 cfm) per person. The office building has ten floors with two outdoor air handling units for each floor, a central hydronic heating and cooling plant, and an economizer. Both energy and economic impacts are presented for five U.S. cities (Miami, Atlanta, Washington, D.C., New York, and Chicago). Reported energy savings ranged from less than 1 % to 3 % for electricity and from 16 % to 22 % for gas. Payback periods of 1.5 years to 2.2 years were estimated for all cities.

In a recent study with a focus on humid climates, Shirey and Rengarajan (1996) simulated the impact of a CO_2-based DCV system in a 400 m^2 (4000 ft^2) office located in Miami, Orlando, and Jacksonville to examine the impacts of ASHRAE Standard 62-1989 ventilation rates on indoor humidity levels. The baseline system, a conventional direct expansion (DX) air-conditioning system with a sensible heat ratio (SHR) of 0.78, was unable to keep the indoor humidity below the target of 60 % relative humidity (RH) when the ventilation rate was increased from 2.5 L/s to 10 L/s (5 cfm to 20 cfm) per person. System modifications considered included a low-SHR DX air-conditioner, a high efficiency low-SHR air-conditioner, a conventional air-conditioner with CO_2-based DCV, a conventional air-conditioner with an enthalpy recovery wheel, a heat pipe assisted air-conditioner, and a conventional air-conditioner with a separate 100 % outdoor air DX unit. The operation of the DCV system was simulated by matching ventilation rates to occupancy profiles. Four alternative systems (DCV, enthalpy wheel, heat pipe, and 100 % outdoor air DX unit) maintained acceptable humidity levels for greater than 97 % of occupied hours. Of the systems with acceptable humidity performance, only the DCV and enthalpy wheel options did so with less than 5 % increases in annual HVAC energy use compared to the conventional system with a ventilation rate of 2.5 L/s (5 cfm) per person. The DCV system also significantly lowered the peak heating demand in Orlando and Jacksonville. An economic analysis showed that the DCV system resulted in annual HVAC operating cost increases of 7 % or less, first cost increases of about 14 %, and life cycle cost increases of about 12 % compared to the system with 2.5 L/s (5 cfm) per person. A case with high internal loads was also examined, with the DCV and enthalpy wheel systems again resulting in the best performance for the smallest increases in cost.

In a recent follow-up study, Davanagere et al. (1997) applied the same methodology with many of the same assumptions as Shirey and Rengarajan (1996) to study HVAC system options including CO_2-based DCV in a Florida school. As in the previous study, the baseline for comparisons was a conventional system with ventilation as required by ASHRAE Standard 62-1981. In addition to DCV, the options simulated included the conventional system with ASHRAE Standard 62-1989 ventilation rates and various combinations of pretreating outdoor air, thermal energy storage, enthalpy recovery wheels, gas-fired desiccant systems, and cold air distribution systems. Results reported included energy use, humidity levels, first costs and life-cycle costs. In general, the DCV system resulted in the smallest or close to the smallest increases in energy costs and installed first costs compared to the baseline system. The thermal energy storage system options generally resulted in the smallest increases (or even decreases) in peak cooling demands and life-cycle costs. DCV was the only option that reduced peak heating demands. Although the DCV system reduced humidity levels compared to the baseline system, many of the other simulated options controlled humidity better.

Nakahara (1996) also discusses a simulation of DCV in a school building with an emphasis on multiple zones and the potential benefit of zoning the ventilation system based on the level of CO_2 demand instead of based on room position. However, little detail is provided on the model, and the baseline for the resulting potential thermal load reduction of 46 % is not clearly defined.

In addition to offices and schools, public spaces have also been the subject of DCV simulation studies. Warren and Harper (1991) evaluated the potential heating energy savings for a CO_2-based DCV system applied to an auditorium in London. Energy simulations were performed using a building energy analysis program (Clarke and McLean 1986) with ventilation rates calculated separately based on occupancy profiles. Assumptions included CO_2 generation of 4.7×10^{-6} m^3/s (1.7×10^{-4} ft^3/s) per person, auditorium volume of 11,150 m^3 (406,000 ft^3), high CO_2 setpoint of 1800 mg/m^3 (1000 ppm(v)), peak daily occupancy of 629, and infiltration rate of 0.4 h^{-1}. Three ventilation scenarios were compared including 100 % outdoor airflow at a rate of 5,020 L/s (10,000 cfm), DCV with a minimum outdoor airflow rate of 3,770 L/s (7,500 cfm), and DCV with no minimum. The DCV with minimum outdoor airflow rate rarely exceeded the minimum rate to maintain CO_2 concentrations below 1800 mg/m^3 (1000 ppm(v)) and saved 26.4 % in heating energy use compared to the 100 % outdoor airflow case. The DCV with no minimum saved 53.3 %.

Ogasawara et al. (1979) evaluated the potential energy savings for a DCV system in a 30,000 m^2 (320,000 ft^2) department store in Tokyo, Japan. Three ventilation strategies were compared including fixed outdoor air at design rate, manual control with maximum ventilation on Sundays (the busiest day) and half of that on weekdays, and DCV. The DCV algorithm used was proportional control with a closed damper at 1440 mg/m^3 (800 ppm(v)) and a fully open damper at 1800 mg/m^3 (1000 ppm(v)). Infiltration assumptions were not specified. Energy use was calculated for 4 cooling months and 4 heating months. The DCV system reduced energy use by 40 % for the cooling season and by 30 % for the heating season. An economic analysis showed an advantage for the DCV system.

Feher and Ambs (1997) reported a study in which measurements of CO_2 concentrations in a school building were used to estimate occupancy and to simulate operation of a DCV system. Four independent zones of the school building, which had recently had the outdoor airflow rate increased above design by 50 %, were included in the energy simulations. PID control was simulated although it was concluded that there was little additional benefit compared to proportional control. No infiltration, interzonal airflow, or air change effectiveness parameters were included in the model. A minimum outdoor airflow of 1 h^{-1} was provided. Annual HVAC energy savings compared to the original design rates were estimated to range from 3 % for the classroom zone up to 17 % for the auditorium zone depending on the control approach.

In a very unique application, Dounis et al. (1996) investigated the potential application of CO_2-based DCV to control ventilation rates for a building with natural ventilation. Simulations were performed in which window opening was adjusted based on measured CO_2 concentrations. Due to concerns over the constant variation of natural ventilation driving forces, fuzzy logic was used instead of conventional on-off or PID control. Carbon dioxide concentrations, window openings, and air temperatures are presented for a simulated day. Although performance was not as good as expected, the authors conclude that the feasibility of such a system was demonstrated.

The simulation case studies reviewed indicated energy savings for DCV systems between 4 % and over 50 % compared to ASHRAE Standard 62-1989 or other design ventilation rates. The energy savings varied widely depending on type of building, control algorithm, building location, assumed occupancy and other assumptions. No parametric or sensitivity analysis has been performed to determine which variables have the most influence on potential energy savings. Also, energy savings are reported with respect to different baseline cases in the different studies. A small number of the studies examined peak demand, economic impacts, humidity and concentrations of other pollutants. These studies verified the concern for increased concentrations of non-occupant generated pollutants, and one study examined potential solutions including scheduled purges. Shortcomings of most of the studies included inadequate treatment of infiltration and interzone airflows and control algorithms.

3.3 Sensor Performance and Location

The performance of a CO_2-based DCV system will clearly depend on the measured CO_2 concentration as reported by the system sensors. Key issues related to these sensors are their accuracy, reliability, and location in the building. This section discusses the research that has been done on sensor performance and location.

Sensor Performance

In the most extensive report on sensor performance, Fahlen et al. (1991 and 1992) describe an evaluation of the performance characteristics of two CO_2, nine humidity, and five mixed-gas sensors in both lab tests and long term field tests. The lab tests consisted of both performance and environmental tests, while the field tests consisted of a repeat of the performance tests after the sensors had been installed in the field for 11 months. The CO_2 sensors displayed acceptable performance for control purposes with a deviation of less than 50 mg/m^3 (30 ppm(v)) at a level of 1800 mg/m^3 (1000 ppm(v)). However, the following problems were identified: time-consuming calibration, sensitivity to humidity, and cross-sensitivity to voltage, temperature and tobacco smoke. Characteristic curves comparing the sensor performance before and after the field trial are presented. At 1800 mg/m^3 (1000 ppm(v)), the deviation from the original result was between 0 mg/m^3 (ppm(v)) and 180 mg/m^3 (100 ppm(v)).

Meier (1993) reports on the performance of two CO_2 and 17 mixed-gas sensors in five different facilities at the University of Zurich. Measurements of CO_2, air quality units (AQU), and occupancy are presented for one day in a restaurant. It is concluded that both mixed-gas and CO_2 sensors are suitable for registering the occupancy level in the restaurant and can provide the reference variable for DCV. The results of the mixed-gas sensors and CO_2 sensors are compared, but no conclusion is reached as to which sensor type is more suitable.

Recently, Okamoto et al. (1996) described the development and field testing of a CO_2 sensor employing solid-state electrolyte technology. The sensor is stated as having an accuracy of

± 20 % and acceptable sensitivity to temperature, humidity, and miscellaneous gases. However, the basis of the statements (i.e., laboratory test results) is not presented. Limited field tests of the sensors in a school and two conference rooms are described. In these tests, the sensors were used as monitors with low, medium and high setpoints of 1260 mg/m^3 (700 ppm(v)), 2520 mg/m^3 (1400 ppm(v)), and 4500 mg/m^3 (2500 ppm(v)) but were not used to control the ventilation system directly.

Several other reports contain more limited discussion of CO_2 sensor performance. The literature review by Raatschen (1990) describes the various types of sensors available. The CO_2 sensors discussed use infrared absorption and are available as two types - photoacoustic and photometric. No actual performance tests were conducted, but a summary of manufacturers' data is provided. Houghton (1995) also describes available sensor types; manufacturer's specifications are presented for five sensors available in the U.S. Issues of accuracy, drift, and temperature and pressure sensitivity are also addressed, although no independent performance tests are reported. Helenelund (1993) also discusses the various sensor options available for DCV systems but does not report on their performance. Based on other published reports, interviews and obtained test results, the suitability of various sensors for different types of facilities is presented from the point of view of both technological and economical performance. In a field test, Sodergren (1982) reported that the sensor calibration drifted from 180 mg/m^3 (100 ppm(v)) to 270 mg/m^3 (150 ppm(v)) during the study. In another field test, Ruud et al. (1991) found that one CO_2 sensor had to be connected to the supply voltage for several days before the output signal became stable.

Sensor Location

In an experimental study aimed at determining the proper location for DCV sensors within a room, Stymne et al. (1990) investigated the dispersion of CO_2 from simulated people in a four-room test house. The following design factors were discussed: transfer of CO_2 from the sources to different locations (referred to as transfer probability), the expected equilibrium concentration at a location, the rate constant of approaching equilibrium from a nonequilibrium state, and concentration fluctuations. The total ventilation flow rate to the test house was varied between two levels with the fraction to each room remaining constant. People were simulated by metallic bodies heated by a 100 W lamp and emitting 0.0069 L/s (0.015 cfm) of CO_2 mixed with prewarmed air. Measurements were taken at 19 locations. Tracer gas measurements were also performed. The measurements showed that good mixing was achieved in rooms with closed doors, and therefore the sensor location is not critical. However, if a room is connected to other spaces by open doors, large differences and instabilities in the CO_2 concentration may occur. The distribution pattern of the tracer gas was similarly nonuniform, indicating that the cause of the distribution pattern is air movement through open doorways and its interaction with air movement from the heated bodies, radiators, cold external walls, and the jet from the inlet duct. It is recommended to place the DCV sensor at mid-height in a room and away from doorways, radiators, windows, people and air inlet devices if possible. It is also recommended that the DCV system have a large time constant in order not to react to the fluctuations in concentration due to nonuniform distribution patterns.

In a follow-up study, Stymne et al. (1991) investigated the CO_2 distribution pattern in an office room with a displacement ventilation system. People were simulated by heated dummies emitting tracer gas. Graphs of iso-concentration contours are presented for several cases. The lack of normal disturbances such as body movements, breathing, heat sources, lighting, and solar heat gain is mentioned as a limitation of the study. It is shown that pollutants emitted from the 'people' are transported to the upper mixed zone in the room and that pollutants emitted at a small heat source or near the wall accumulate below the interface between the upper and lower zones. The interface is displaced about 0.2 m (0.66 ft) upwards around the heated bodies, ensuring the occupants better air quality than the surrounding air, even if they are above the interface. A test with a mixing ventilation system showed a similar plume above the heated dummies but no stratification outside the plume. It is concluded that DCV in a displacement ventilated room is a suitable means of controlling the level of the interface between the uncontaminated air in the upper zone and the polluted air in the lower zone. The sensors should be located at the height of the occupants' heads. Also, the setpoint should be lower than usual, for example below 1440 mg/m^3 (800 ppm(v)), so that the DCV system will be activated.

A common alternative to locating DCV sensors in individual rooms is to locate them in the ventilation system return ductwork. Reardon and Shaw (1993) and Reardon et al. (1994) compared CO_2 concentrations in the central return air shafts, individual floor return intakes, and occupied space in a 22-story office building. Measurements showed that the individual floor return grilles represented the spatial average concentrations in the occupied space, and that the measurements at the top of the central return shafts represented the concentrations at the floor return intakes. Therefore, it was concluded that the top of the return shafts is an appropriate location for the sensors of a DCV system. However, the setpoint should be adjusted (lowered) to account for variability in the occupied zones to avoid high local exposures.

Bearg (1994) also compares the merits of single and multiple point DCV systems. A system is described with multiple sampling points and a single detector installed in a 5-story building. In addition to operating the DCV system, advantages credited to the multipoint system include identifying both leakages in the system and episodes of increased outdoor contamination such as vehicle exhaust at a loading dock. Also, the use of a single detector ensures that differences in measured concentrations for different sampling points are not due to calibration differences. Such a system could also be automatically recalibrated with a known CO_2 concentration. Houghton (1995) discusses this multipoint system including its accuracy and automatic calibration advantages. However, the system is claimed to be more costly than a system with multiple detectors and a central computer. Some data collected by the multipoint system is presented.

Several other reports briefly discuss sensor location issues. In another field test, Sodergren (1982) presented graphs of the CO_2 concentration at multiple locations in an office but did not make specific recommendations on sensor location. In a test in a conference room, Ruud et al. (1991) found that concentrations measured at the wall and in the exhaust air were nearly identical with the wall-mounted sensor having a 2-min delay compared to the exhaust air. In a simulation study of a DCV system applied to an office building with floors having different occupant densities, Enermodal (1995) found that a system with sensors in the return duct of each floor had little impact on IAQ and energy use compared to a system with a sensor in the central return. Installing sensors in each zone ensured CO_2 concentrations below 1800 mg/m^3 (1000 ppm(v)) (the setpoint) in all zones and increased energy use slightly, but at a higher installation cost due to the additional sensors. Central control with a setpoint of 1440 mg/m^3 (800 ppm(v)) offered similar performance to individual zone control with a setpoint of 1800 mg/m^3 (1000 ppm(v)), but at a much lower installation cost.

Although many DCV studies have touched on the subjects of sensor performance and location, only a few have examined these issues in detail. In general, sensor performance characteristics have been found to be adequate for controlling a DCV system although concerns about calibration and sensitivity to humidity and temperature have been expressed. Sensor calibration concerns are being addressed by either use of a second detector tuned to a wavelength that isn't absorbed by CO_2 to provide a reference value to correct for sensor drift over time or "self-calibrating" by checking the CO_2 level at night when indoor concentrations are expected to drop to outdoor levels (Schell and Int-Hout 2001). Contradicting opinions on sensor location have been expressed with some studies advocating a system with a single central measurement in the HVAC return system and others preferring a system with multiple measurement points.

3.4 Application

In addition to the studies of the performance of CO_2-based DCV systems, there have also been a growing number of reports that describe how to apply these systems. These reports range from general descriptions of CO_2-based DCV to detailed descriptions of control algorithms. This section reviews a number of these reports.

One of the earliest discussions of using CO_2 to control outdoor air intake as a means of saving energy was presented by Kusuda (1976). This paper presented some of the theoretical background of how indoor CO_2 concentrations vary as a ventilation system is turned on and off. Sample calculations showed potential energy savings of 40 % for an office space. Another early discussion of the energy savings potential of CO_2 control was presented by Turiel et al. (1979). This paper discussed a number of DCV control options including water vapor and concluded that CO_2 control appeared to be the most satisfactory approach.

Recently, one of the more detailed discussions of the application of DCV was reported by Schell et al. 1998. DCV topics covered include potential energy savings with DCV, determining locations for CO_2 sensors, control strategies (including setpoint, proportional, and exponential or PID), consideration of outdoor levels of CO_2, estimation of building ventilation rates using CO_2, models for selection of DCV strategy, and benefits of DCV. Additionally, Schell et al. discuss applying CO_2-based DCV in compliance with ASHRAE Standard 62-1989 (ASHRAE 1990).

A general discussion of the principles of DCV in office buildings is presented by Davidge (1991) and Houghton (1995). These papers discuss the circumstances under which DCV might be expected to be most effective including the existence of unpredictable variations in occupancy, a building and climate where heating or cooling is required for most of the year, and low pollutant emissions from non-occupant sources. Davidge points out that when such a system is considered, one must address the base ventilation rate that is not controlled by DCV in order to control these non-occupant pollutant sources. The impact of free cooling on DCV systems is also discussed, noting that long periods of free cooling will reduce the potential energy savings. The potential for purge ventilation, both before and after occupancy, to control non-occupant sources is also discussed.

Similar discussions of the application of CO_2-based DCV are presented by Houghton (1995) and in an application guide published by Telaire (n.d.). These publications contain background information on CO_2 control of ventilation and describe the potential energy savings benefits. Strategies for the use of CO_2-based DCV are also described including simple setpoint control where the outdoor air intake damper is either open or closed depending on the indoor CO_2 concentration, proportional control in which the intake damper or outdoor air fan flow is proportional to the CO_2 concentration, and PID (proportional-integral-derivative) control which considers the rate of change in the CO_2 concentration. Recommendations are made on the application of these techniques based on the occupancy level.

Descriptions of specific control algorithms are presented by Vaculik and Plett (1993), Federspiel (1996), Bjorsell (1996), and the Telaire application guide (n.d.). In their paper, Vaculik and Plett discuss the principles of CO_2-based DCV including setpoint and proportional control. They then describe a control approach that accounts for differences between CO_2 concentration at the measurement location and the critical location in the building and in which the control setpoint is adjusted to account for differences between the measured concentration and the setpoint.

Federspiel (1996) also reports on a control algorithm, referred to as On-Demand Ventilation Control (ODVC), and presents a simple simulation to demonstrate its performance. The ODVC strategy attempts to set the ventilation rate proportional to the occupant density even under transient conditions by using a well-mixed single zone model to estimate the current CO_2 generation rate from measured concentrations and airflows. A simple example is presented to show the ODVC strategy controls the CO_2 concentration below 1800 mg/m^3 (1000 ppm(v)) by reacting quickly to a step change in occupancy, while a strategy of PI control of measured CO_2 concentration allows CO_2 to overshoot the setpoint value. Issues regarding the impact on energy use and the potential effect of well-mixed single zone model inadequacies are not addressed. Ke and Mumma (1997) and Wang and Jin (1998) describe similar algorithms.

Bjorsell (1996) also focuses on the description and simulation of a DCV control algorithm, presenting a simple simulation example. The control algorithm, called Linear Quadratic, attempts to calculate the optimal system flow to minimize a cost function that depends on concentration and ventilation flow. However, the cost function is not specified and, although the control method may be optimal with respect to a given cost function, it also depends on all physical data being known and may not be practical to implement.

As mentioned earlier in the sections on field and simulation cases studies, a variety of control setpoints have been used, and many descriptions of the application of CO_2 control contain only limited discussion of how to determine the appropriate setpoint. Schultz and Krafthefer (1993) present a method for determining a CO_2 setpoint based on the Indoor Air Quality procedure in ASHRAE Standard 62. This method employs a two-zone model of the ventilated space and considers the ventilation efficiency of the space. Nomographs are presented for use in determining the CO_2 setpoints.

The use of CO_2 control of outdoor air is discussed relative to other approaches of outdoor air control in papers by Elovitz (1995) and by Janu et al. (1995). Elovitz discusses various options for controlling minimum outdoor air intake rates in VAV systems including: sequencing supply and return fans; controlling return or relief fans based on building pressure; measuring outdoor air intake rates directly; fan tracking; controlling the pressure in the intake plenum; outdoor air injection fans; and, CO_2 control. Advantages and disadvantages of each approach are discussed. Elovitz points out that CO_2 control does not necessarily assure satisfactory indoor air quality, depending on the existence and strength of contaminant sources that are not proportional to the number of occupants. Janu et al. (1995) discuss some of the same methods of outdoor airflow control and raise the same cautions regarding CO_2 control and non-occupant contaminant sources.

In addition to a general discussion of DCV, Meier (1995) reports a sensitivity analysis on parameters affecting the payback period for modifying a conventional ventilation system to add DCV capability. Although few details of the calculation are presented, the total airflow rate is reported as the most significant parameter determining payback period. However, operating hours were also found to be significant. More recently, Meier (1998) provided estimated potential energy-cost savings for a range of DCV applications based on case studies and experiences of control companies. These estimates and additional ones from Mansson (1994) are presented in Table 2. Mansson provides background information on CO_2 DCV systems, discusses strategies for base and variable ventilation rates based on application type, and presents a six-step flowchart for determining the feasibility of DCV for an application. As expected, the energy savings in Table 2 are largest for high density spaces with generally variable occupancy, such as the various halls, theatres and cinemas. The lowest savings are seen in the office spaces, which generally have lower occupancy densities with less variation than the other spaces.

Application	Energy-cost savings range
Schools	20 % to 40 %
Day nurseries	20 % to 30 %
Restaurants, canteens	20 % to 50 %
Lecture halls	20 % to 50 %
Open-plan offices (40 % average occupancy)	20 % to 30 %
Open-plan offices (90 % average occupancy)	3 % to 5 %
Entrance halls, booking halls, airport check-in areas	20 % to 60 %
Exhibition halls, sports halls	40 % to 70 %
Assembly halls, theatres, cinemas	20 % to 60 %

Table 2 Estimated energy-cost savings from DCV (Meier 1998 and Mansson 1994)

3.5 Summary and Conclusions

This literature review has described the research into the application of CO_2-based DCV. It has covered case studies conducted in the field and through computer simulation, research on sensors, and discussions of the application of CO_2 control. This section summarizes a number of findings of the literature review and identifies research needs. Table 3 summarizes the literature reviewed in terms of the type of report and topics addressed.

There is fairly wide consensus on the best applications for CO_2 control. Most discussions of CO_2-based DCV mention the following building types as good candidates: public buildings such as cinemas, theaters and auditoria, educational facilities such as classrooms and lecture halls, meeting rooms, and retail and restaurant establishments. However, it is interesting to note that most of the case studies have investigated office buildings. As presented by Davidge (1991), the following building features correspond to situations where CO_2-based DCV are most likely to be effective:

- the existence of unpredictable variations in occupancy
- a building and climate where heating or cooling is required for most of the year
- low pollutant emissions from non-occupant sources.

There have been a number of valuable demonstration projects in real buildings, and many of these have shown significant energy savings through the use of CO_2 control. However, several cases exist where the indoor CO_2 concentration was rarely high enough for the outdoor air intake dampers to open, suggesting a mismatch between building occupancy, ventilation rates, and control algorithms and setpoints. A significant shortcoming of several of the field tests, as well as of the computer simulation studies, was the inclusion of little or no description of the CO_2 sensors or control algorithm investigated in the study. These omissions make it difficult to evaluate which approaches work best and under what circumstances.

While CO_2 DCV can control occupant-generated contaminants effectively, it may not control contaminants with non-occupant sources as well. The control of such non-occupant sources, such as some building materials and outdoor air pollution, is a difficult issue because one cannot engineer for these sources unless their source strengths and indoor concentration limits are known. However, this information is not readily available for most contaminants and sources. A practical solution is to maintain a base ventilation rate at all times, which can be proportional to floor area. A morning purge with outdoor air may also be a good strategy for controlling the buildup of these contaminants over night, and it may be equally applicable to non-DCV systems. An outdoor air purge cycle during the day is another option for controlling non-occupant sources.

The research on sensors indicates that currently available technology is adequate for use in these systems. Some questions have been raised regarding calibration frequency, drift, and temperature effects but new calibration methods have been developed to address these concerns. There is still some debate regarding sensor location, in particular whether to use a single sensor centrally located in the system return or multiple sensors located in the returns for whole floors or in critical spaces, such as conference rooms. Whenever a central location is suggested, the issue of variability among spaces is almost always mentioned. Using a lower setpoint with a central sensor is often suggested as one means of dealing with the variability issue.

A number of needs for more research and information were identified in this literature review. For example, more system-specific guidance on application of CO_2-based DCV is needed. This guidance should be based on system type, zoning, and expected variations in occupancy patterns among the zones. The factors that impact energy savings and other performance issues are becoming better understood, but more sensitivity analysis would be helpful. As mentioned in the literature review, it would still be extremely useful to investigate more CO_2-based DCV installations and document them in terms of design and performance. Another issue meriting attention are the positive benefits of using CO_2-based DCV to maintain ventilation rates at design levels that help to guarantee sufficient ventilation to occupants, as opposed to its use to using CO_2 control to reduce ventilation rates.

	Field test	Simulation	Sensor performance	Sensor location	Application	Energy	IAQ	Economic	Control algorithm	Office	School	Conference room	Other
Anon. 1986	X					X							X
Bearg 1994				X	X								
Bjorsell 1996		X							X				
Brandemuehl and Braun 1999		X				X				X	X		X
Carpenter 1996/Enermodal 1995		X		X		X	X		X	X			
Chan et al. 1999		X					X						X
Davanagere et al. 1997		X				X	X	X			X		
Davidge 1991	X					X	X			X		X	
Donnini et al. 1991 and Haghighat and Donnini 1992	X					X	X	X		X			
Dounis et al. 1996		X							X	X			
Elovitz 1995					X								
Emmerich et al. 1994		X				X	X			X		X	
Fahlen et al. 1991 and 1992			X	X									
Federspiel 1996		X							X			X	
Feher and Ambs 1997		X				X			X		X		
Fehlmann et al. 1993	X					X	X						X
Fleury 1992	X						X					X	
Gabel et al. 1986	X					X	X			X			
Haghighat et al. 1993 and Zmeureanu and Haghighat 1995		X				X		X		X			
Helenelund 1993					X								
Houghton 1995			X		X								
Huze et al. 1994	X											X	
Janssen et al. 1982 and Woods et al.1982	X					X	X				X		
Janu et al. 1995					X								
Ke and Mumma 1997									X				
Knoespel et al. 1991		X				X	X			X		X	
Kulmala et al. 1984	X					X							X
Kusuda 1976		X			X	X				X			
Mansson 1994					X	X							
Meckler 1994		X				X		X		X			
Meier 1993			X										
Meier 1995 and 1998					X	X	X	X	X	X			X
Nakahara 1996		X				X					X		
Ogasawara 1979		X				X		X					X
Okamoto et al. 1996			X										
Potter and Booth 1994	X				X								X
Raatschen 1990			X		X								
Reardon and Shaw 1993 and Reardon et al. 1994				X									
Ruud et al. 1991	X			X								X	
Schell et al. 1998 and Schell and Int-Hout 2001			X		X	X	X		X				
Schultz and Krafthefer 1993					X								
Shirey and Rengarajan 1996		X				X	X	X		X			
Sodergren 1982	X		X	X			X			X			
Sorensen 1996		X				X			X	X		X	
Strindehag et al. 1990	X									X	X	X	
Stymne et al. 1990 and Stymne et al. 1991				X	X								
Telaire Systems, Inc					X				X				
Turiel et al. 1979					X	X							
Vaculik and Plett 1993				X	X				X				
Wang and Jin 1998		X				X	X		X	X		X	
Warren 1982	X					X							X
Warren and Harper 1991		X				X							X
Zamboni et al. 1991	X					X	X						X

Table 3 Summary of Literature Review

4. TECHNOLOGY UPDATE: SENSORS

The only technologies unique to the application of carbon dioxide demand controlled ventilation are the CO_2 sensors themselves. The remaining control hardware and software (including algorithms) are common with other HVAC control applications. This section covers only the CO_2 sensors, describing the available technology and some of the relevant performance issues.

Most major HVAC equipment manufacturers offer CO_2 demand controlled ventilation as an option, though some highlight its use more than others. However, their application literature does not generally focus on the CO_2 sensing technology. Presumably these HVAC manufacturers are using sensors manufactured by other firms.

Sensor manufacturers are definitely promoting the use of CO_2 DCV and several of them provide a good deal of technical information on the performance and application of their products. However, there has not been a great deal of information published on CO_2 sensing for control. A recent article by Schell and Int-Hout (2001) provides a brief description of the sensing technology. In addition, the International Energy Agency effort (Annex 18) that studied demand control ventilation in general did perform limited testing of CO_2 sensors (Fahlen et al. 1992).

There are basically two types of CO_2 sensors used for ventilation control, photometric and photoacoustic. Photometric sensors contain a light source that emits in the infrared range and an optical filter that ensures only wavelengths in the absorbing spectrum of CO_2 enter the cell containing the air sample. A photodetector measures the light intensity at a wavelength that is absorbed by CO_2. The higher the CO_2 concentration in the sample air, the lower the measured light intensity. Photoacoustic sensors also employ an infrared light source with an optical filter. The CO_2 molecules in the cell absorb the infrared energy, which in turn increases the molecular vibration and generates an acoustic field. A microphone picks up this field and converts it to an electronic signal related to the CO_2 concentration.

Some of the issues affecting sensor performance include interference from other gases (e.g. water vapor), accuracy and drift. The various sensors on the market address these issues using different strategies. Sensor drift arises due to aging of the light source or particle/dust buildup on the optical components. Some sensors use a second detector tuned to a wavelength that isn't absorbed by CO_2 to provide a reference value to correct for sensor drift over time. Another approach is to protect the sensor with a gas permeable membrane to avoid contamination by dust. Photoacoustic sensors are not as sensitive to dirt and dust, but are still subject to aging of the light source. Some sensors check the CO_2 level at night when indoor concentrations are expected to drop to outdoors and "self-calibrate" to account for drift.

The IEA Annex 18 sensor testing effort included testing of only two CO_2 sensors, one photometric and another photoacoustic (Fahlen et al. 1992). The testing involved sensor response as well as the impacts of temperature, mechanical vibration and electrical noise. While the testing was limited to only two sensors, the study concluded that these sensors showed acceptable performance for control but noted that calibration could be a time consuming process based on the sensor rise times of around 10 min.

Sensor manufacturers are continually working to develop cheaper, more accurate and more stable sensors. Their cost is driven to a large degree by the number of sensors that are manufactured and sold. In 1998, prices ranged from around \$400 to \$500 per sensor, but have decreased by about 50 % based on continued use (Schell and Int-Hout 2001). This trend is likely to continue as their use expands further.

5. STANDARDS AND REGULATORY CONTEXT

Carbon dioxide demand controlled ventilation clearly must be applied in a manner that is consistent with the relevant building codes and standards. This section discusses the standards and regulatory context relevant to CO_2 DCV, specifically ASHRAE Standard 62 and the California Energy Efficiency Standards.

ASHRAE Standard 62

Before discussing CO_2 DCV in the context of Standard 62, it is appropriate to address some confusion that exists regarding the standard and indoor CO_2 levels. For example, a common misunderstanding exists that if indoor carbon dioxide concentrations in a building are maintained below 1800 mg/m^3 (1000 ppm(v)) or within 1260 mg/m^3 (700 ppm(v)) of outdoors, then the building is in compliance with ASHRAE Standard 62. ASHRAE Standard 62 contains two paths to compliance, the Ventilation Rate Procedure and the Indoor Air Quality Procedure. The Ventilation Rate Procedure requires that one determine the design ventilation rate of a building based on the space-use in the building, the number of occupants and the outdoor air requirements for various space-use categories in Table 2 of the standard. The Ventilation Rate Procedure also contains requirements for contaminant levels in the outdoor air and that no unusual contaminants or sources exist. While compliance with the Ventilation Rate Procedure is likely to maintain indoor carbon dioxide concentrations within 1260 mg/m^3 (700 ppm(v)) of outdoors (corresponding to about 1800 mg/m^3 (1000 ppm(v)) for typical outdoor CO_2 concentrations), the other requirements of the procedure must also be met to achieve compliance with the entire standard.

The Indoor Air Quality Procedure of the 1989 version of the standard contained a guideline for indoor carbon dioxide concentrations of 1800 mg/m^3 (1000 ppm(v)), but that guideline was removed in the 1999 version. However, complying with this guideline alone was never sufficient for achieving compliance with the standard. In addition to this carbon dioxide guideline, the IAQ Procedure also contained and still contains limits for four other contaminants of predominantly outdoor origin in Table 1 of the standard and three others in Table 3. In addition, one also must to keep all other known contaminants of concern below specific levels. The Indoor Air Quality Procedure also contains a requirement for the subjective evaluation of the acceptability of the level of those contaminants for which no objective measures of acceptability are available. While it may not be clear how one identifies these contaminants of concern and the associated levels of acceptability, it is clear that simply maintaining carbon dioxide below 1800 mg/m^3 (1000 ppm(v)) is not sufficient.

That being said, let us now review how the standard does address the issue of CO_2 demand controlled ventilation. Section 6 of this standard, Procedures, provides the means of designing building ventilation systems for achieving acceptable indoor air quality. Two procedures exist for doing so, the Ventilation Rate Procedure and the Indoor Air Quality (IAQ) Procedure. The former prescribes minimum outdoor air ventilation requirements for a number of different space types, expressed as (L/s) cfm per person, (L/s-m^2) cfm/ft^2 of floor area or cfm (L/s) per room, with the units depending on the space type. The standard does not specifically discuss the application of demand-controlled ventilation, however, it is quite logical to apply this approach to spaces where the outdoor air requirement is expressed as (L/s) cfm per person. If carbon dioxide, or some other demand control approach, is employed as a "people counter," then the outdoor air could be varied in response to changes in occupancy. In fact, ASHRAE has issued interpretation #33 of Standard 62-1999 (also referred to as interpretation #27 of Standard 62-1989) that allows this application. The interpretation allows the use of CO_2-DCV as long as other provisions of the standard (specifically requirements related to intermittent occupancy) have not been used to reduce the estimated occupancy, CO_2 is not being removed by other methods such as air cleaning, and a control algorithm is used to achieve the rates in Table 2 of the standard. The interpretation does specifically allow for a number of different control algorithms, including " make or break" (on/off), proportional, proportional-integral, and proportional-integral-derivative, and specifically mentions the use of the difference between indoor and outdoor CO_2 levels in these controls. However, the interpretation notes " good practice and the rationale on which the ventilation rates in Table 2 are based, indicates the need for a non-zero base ventilation rate to handle non-occupant sources whenever the space is occupied." Therefore, some residual ventilation needs to be provided to handle non-occupant contaminant sources, but neither the standard nor the interpretation indicates how much ventilation that is.

The IAQ Procedure is a performance-based method for providing acceptable IAQ in which the design is based on the control of certain "contaminants of concern" to specified acceptable levels. The standard neither identifies the contaminants on which to base the design nor the acceptable levels; that is up to the user of the standard. Carbon dioxide DCV could be one means of implementing the IAQ Procedure, but realistically one would also need to address contaminants that are not generated at rates associated with the number of occupants.

Another aspect of the standard that is relevant to CO_2-DCV is the indoor CO_2 guideline that was in the 1989 version of the standard. Table 3 of the standard (both the 1989 and 1999 versions) contains guidelines for selected air contaminants for potential use with the IAQ Procedure. The 1989 version of Table 3 included a limit of 1800 mg/m^3 (1000 ppm(v)) for indoor CO_2, which was the subject of much confusion. Specifically, some readers of the standard interpreted this guideline as indicating that indoor CO_2 levels about 1800 mg/m^3 (1000 ppm(v)) were a health hazard. In fact, this guideline was based on the association with indoor CO_2 levels with the level of odor due to human bioeffluents. Since then, the approval of Addendum 62f to the standard in 1999 and its incorporation in Standard 62-1999 removed the 1800 mg/m^3 (1000 ppm(v)) CO_2 guideline and replaced it with the actual contaminant of interest, i.e., human bioeffluents. Additional changes to Appendix D of the standard explained that odor levels from human bioeffluents were likely to be acceptable to the majority of visitors entering a space if the indoor CO_2 level were no more than 1260 mg/m^3 (700 ppm(v)) above outdoors.

California's Energy Efficiency Standards for Residential and Nonresidential Buildings

The CEC standards (1999), often referred to as Title 24, discuss demand-controlled ventilation under Section 121 - Requirements for Ventilation. In addition to providing minimum outdoor air ventilation rates, section 121 (c) discusses operation and control of outdoor air. An exception to the requirement that the specified outdoor air rates shall be supplied whenever the space is occupied states that the outdoor air rate may be reduced to 0.76 L/s-m^2 (0.15 cfm/ft^2) if the system is controlled by an approved demand controlled ventilation device and in the case of CO_2 control, the indoor CO_2 level is limited to no more than 1440 mg/m^3 (800 ppm(v)) while the space is occupied. The basis for this limit is not provided in the document, but it should be noted that an indoor CO_2 level of 1440 mg/m^3 (800 ppm(v)) corresponds to about 11.5 L/s (23 cfm) per person of outdoor air under steady state conditions.

These standards were revised in January 2001 in a document containing emergency regulations, referred to as AB 970 (CEC 2001a). The requirements for CO_2 demand-controlled ventilation remained largely the same as the 1999 standards, with a few exceptions. One significant change is that demand control ventilation is required for spaces with fixed seating and occupant densities less than or equal to about 9 m^2 (10 ft^2) per person and for spaces with outdoor air capacities greater than or equal to 1400 L/s (3000 cfm). However, note that these requirements are for any form of demand control that reduce outdoor air intake based on occupancy, not just those based on CO_2 sensing. While the standard is not specific, such control could be based on timers, occupancy sensors and other approaches. When the control device is based on indoor CO_2 levels, the emergency standards still require that the indoor CO_2 levels not exceed 1440 mg/m^3 (800 ppm(v)) when the space is occupied. As noted above, this CO_2 level corresponds to an outdoor air ventilation rate of 11.5 L/s (23 cfm) per person at steady-state. In addition, the 2001 standard requires a sensor in the space or in a return airstream from the space, with one sensor for every 2300 m^2 (25,000 ft^2) of habitable space.

Revisions to the 2001 emergency standards (CEC 2001b) were recently issued for review prior to their adoption on April 4, 2001. These proposed revisions add an exception to the 1440 mg/m^3 (800 ppm(v)) if the ventilation rate is greater than or equal to the rate required by the standard.

The current versions of ASHRAE Standard 62-1999 and California's Title 24 Energy Efficiency Standards both allow the use of CO_2 demand-controlled ventilation. However, neither document provides much application guidance or specific requirements in several important areas including sensors (number, placement, accuracy, calibration or maintenance), control algorithms and setpoints, or base ventilation rates. To effectively realize the energy saving potential of this technology, such guidance is definitely needed.

6. SUMMARY

Carbon dioxide demand controlled ventilation (DCV) attempts to achieve acceptable indoor air quality (IAQ) at reduced energy cost by matching a ventilation system's outdoor airflow rate to the real-time occupancy as indicated by indoor CO_2 levels. The potential advantages of CO_2-based DCV are increased ventilation when occupancy is high, a feedback control mechanism to ensure acceptable IAQ and energy savings from decreased ventilation when occupancy is low. While the energy savings potential of this approach has been highlighted in several studies, there are still some important questions related to the implementation of CO_2-based DCV. One of the most critical issues is that low CO_2 levels alone do not guarantee acceptable IAQ. For example, the concentrations of non-occupant generated pollutants may not be well controlled by such a system, or at least they can become elevated during periods of low occupancy due to decreased ventilation. Also, nonuniformities in air distribution and in building occupancy can present difficulties in locating sensors such that a representative CO_2 concentration is measured. While the potential energy savings have been identified in a number of earlier studies, additional study is needed to help designers estimate the actual saving that can be realized in specific situations. In addition, work remains to be done on improving CO_2 sensors and in developing application guidance. Future phases of this project will focus on increasing our understanding of the energy savings potential and in developing improved application guidance.

7. ACKNOWLEDGEMENTS

This report was supported by the California Energy Commission through the Architectural Energy Corporation. The authors acknowledge the contributions of Vern Smith of the Architectural Energy Corporation, James Braun of Purdue University, William Healy of NIST and Amy Musser of the University of Nebraska.

8. REFERENCES

ACGIH. 2001. 2001 Threshold Limit Values (TLVs) for Chemical Substances and Physical Agents and Biological Exposure Indices (BEIs). American Conference of Governmental Industrial Hygienists.

Anon. 1986. Ventilation Control by Measurement of Carbon Dioxide Levels in Public Entertainment Buildings. Energy Technology Support Unit AERE Harwell, Oxfordshire.

Apte, M.G., W.J. Fisk and J.M. Daisey. 2000. Associations Between Indoor CO_2 Concentrations and Sick Building Syndrome Symptoms in US Office Buildings: An Analysis of the 1994-1996 BASE Study Data. Indoor Air 10 (4): 246-257.

ASHRAE. 1997. Fundamentals Handbook, Chapter 8 Thermal Comfort. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1989. Standard 62-1989, Ventilation for Acceptable Indoor Air Quality. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.,

ASHRAE. 1999. Standard 62-1999, Ventilation for Acceptable Indoor Air Quality. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.,

ASTM. 1998. Standard Guide for Using Indoor Carbon Dioxide Concentrations to Evaluate Indoor Air Quality and Ventilation. American Society for Testing and Materials, West Conshohoken, PA, D6245-98.

Axley, J. 1988. Progress Toward a General Analytical Method for Predicting Indoor Air Pollution in Buildings: Indoor Air Quality Modeling Phase III Report. Gaithersburg, Maryland: National Bureau of Standards, NBSIR 88-3814.

Bearg, D.W. 1994. Second Generation Demand-Controlled Ventilation Systems. Proceedings of ASHRAE IAQ 94: 169-174.

Bell, S.J. and B. Khati. 1983. Indoor Air Quality in Office Buildings. Occupational Health in Ontario. 4: 103-118.

Berg-Munch, B., G. Clausen and P.O. Fanger. 1986. Ventilation Requirements for the Control of Body Odor in Spaces Occupied by Women. Environment International 12: 195-199.

Bjorsell, N. 1996. Control Strategies for Demand Controlled Ventilating Systems. Proceedings of Roomvent ‘96.

Brandemuehl, M.J. and J.E. Braun. 1999. The Impact of Demand-Controlled and Economizer Ventilation Strategies on Energy Use in Buildings. ASHRAE Transactions 105 (2): 39-50.

Bright, P.D., M.J. Mader, D.R. Carpenter and I.Z. Hermon-Cruz. 1992. Guide for Indoor Air Quality Surveys. Armstrong Laboratory, Brooks Air Force Base, Texas, AL-TR-1992-0016.

California Energy Commission. 1999. Energy Efficiency Standards for Residential and Nonresidential Buildings, P400-98-001.

California Energy Commission. 2001a. AB970 Energy Efficiency Standards for Residential and Nonresidential Buildings, P400-01-001.

California Energy Commission. 2001b. AB970 Energy Efficiency Standards for Residential and Nonresidential Buildings, P400-01-009.

Cain, W.S., B.P. Leaderer, R. Isseroff, L.G. Berglund, R.J. Huey, E.D. Lipsitt and D. Perlman. 1983. Ventilation Requirements in Buildings - I. Control of Occupancy Odor and Tobacco Smoke Odor. Atmosphere Environment. 17: 1183-1197.

Carpenter, S.C. 1996. Energy and IAQ Impacts of CO_2-Based Demand-Controlled Ventilation. ASHRAE Transactions 102 (2): 80-88.

CEC. 1992. Guidelines for Ventilation Requirements in buildings. Report No. 11, European Concerted Action Indoor Air Quality & Its Impact on Man. Commission of the European Communities, Brussels,

Chan, G.Y., C.Y. Chao, D.C. Lee, S.W. Chan, and H. Lau. 1999. Development of a Demand Control Strategy in Buildings using Radon and Carbon Dioxide Levels. Proceedings of Indoor Air 99 1:48-53.

Clarke, J. and D. McLean. 1986. ABACUS. E.S.P. Documentation - A Plant and Energy Simulation System. Strathclyde University.

Davanagere, B.S., Shirey, D.B., Rengarajan, K., and F. Colacino. 1997. Mitigating the Impacts of ASHRAE Standard 62-1989 on Florida Schools. ASHRAE Transactions 103 (1): 241-258.

Davidge, B. 1991. Demand Controlled Ventilation Systems in Office Buildings. Proceedings of the 12th AIVC Conference Air Movement & Ventilation Control within Buildings: 157-171. Coventry, Great Britain: Air Infiltration and Ventilation Centre.

Di Giacomo, S.M. 1999. DCV: History, Theory, Myths. Engineered Systems, February 1999.

Donnini, G., F. Haghighat, and V.H. Hguyen. 1991. Ventilation Control of Indoor Air Quality, Thermal Comfort, and Energy Conservation by CO_2 Measurement. Proceedings of the 12th AIVC Conference Air Movement & Ventilation Control within Building: 311-331.

Dounis, A.I., M. Bruant, G. Guarracino, and P. Michel. 1996. Indoor Air-Quality Control by a Fuzzy-Reasoning Machine in Naturally Ventilated Buildings. Applied Energy; 1:11-28.

Elovitz, D.M. 1995. Minimum Outside Air Control Methods for VAV Systems. ASHRAE Transactions 101 (2): 613-618.

Emmerich, S.J., J.W. Mitchell, and W.A. Beckman. 1994. Demand-Controlled Ventilation in a Multi-Zone Office Building. Indoor Environment 3: 331-340.

Emmerich, S.J. and A.K. Persily. 1997. Literature Review on CO_2-Based Demand-Controlled Ventilation. *ASHRAE Transactions* 103 (2): 229-243.

Enermodal. 1995. An Evaluation of the Effect of CO_2-Based Demand-Controlled Ventilation Strategies on Energy Use and Occupant-Source Contaminant Concentrations. ASHRAE Research Project 740-TRP. Waterloo, Ontario: Enermodal Engineering Limited.

Enermodal. 1990. Enerpass Version 3.0. Waterloo, Ontario: Enermodal Engineering Limited.

EPA. 1991. Introduction to Indoor Quality. A Reference Manual. U.S. Environmental Protection Agency, U.S. Public Health Service, National Environmental Health Association, EPA/400/3-91/003.

EPA. 1999. Exposure Factors Handbook. U.S. Environmental Protection Agency, EPA/600/C-99/001.

Fahlen, P., H. Andersson, and S. Ruud. 1991. Demand Controlled Ventilation - Evaluation of Commercially Available Sensors. Proceedings in the 12th AIVC Conference Air Movement & Ventilation Control within Buildings: 201-216.

Fahlen, P., H. Andersson and S. Ruud. 1992. Demand Controlled Ventilating Systems Sensor Tests. Swedish National Testing and Research Institute, Boras, Sweden, SP Report 1992:13.

Fanger, P.O. and B. Berg-Munch. 1983. Ventilation and Body Odor. Engineering Foundation Conference on Management of Atmospheres in Tightly Enclosed Spaces, Santa Barbara, CA. 45-50.

Fanger, P.O. 1988. Introduction of the Olf and Decipol Units to Quantify Air Pollution Perceived by Humans Indoors and Outdoors. Energy and Buildings. 12: 1-6.

Federspiel, C.C. 1996. On-Demand Ventilation Control: A New Approach to Demand-Controlled Ventilation. Proceedings of Indoor Air '96.

Feher, T.R. and L.L. Ambs. 1997. The Use of CO_2 Readings with Fixed Ventilation to Predict Energy Savings with Demand-Controlled Ventilation. Proceedings of Healthy Buildings/IAQ '97.

Fehlmann, J., H. Wanner, and M. Zamboni. 1993. Indoor Air Quality and Energy Consumption with Demand Controlled Ventilation in an Auditorium. Proceedings of the 6th International Conference on Indoor Air Quality and Climate. Vol. 5: 45-50.

Fleury, B. 1992. Demand Controlled Ventilation: A Case Study. Proceedings of the 13th AIVC Conference Ventilation for Energy Efficiency and Optimum Indoor Air Quality: 343-346.

Gabel, S. D., J.E. Janssen, J. O. Christoffel, and S. E. Scarborough. 1986. Carbon Dioxide-Based Ventilation Control System Demonstration. U. S. Department of Energy, DE-AC79-84BP15102.

Haghighat, F. and G. Donnini. 1992. IAQ and Energy-Management by Demand Controlled Ventilation. Environmental Technology. 13: 351-359.

Haghighat, F., R. Zmeureanu, and G. Donnini. 1993. Energy Savings in Building by Demand Controlled Ventilation System. Proceedings of the 6th International Conference on Indoor Air Quality and Climate. 5: 51-56.

Helenelund, C. 1993. Future of DCV - What is Economically Feasible? Proceedings of Air the 6th International Conference on Indoor Air Quality and Climate. 5: 57-62.

Houghton, D. 1995. Demand-Controlled Ventilation: Teaching Buildings to Breathe. Boulder, CO: E. Source, Inc, TU-95-10.

Huze, M.H., Meneboo, F., and J.B. Hoffmann. 1994. Air quality sensors: a field evaluation. Proceedings of Roomvent 94 2:491-501.

Iwashita, G., K. Kimura, S. Tanabe, S. Yoshizawa and K. Ikeda. 1990. Indoor Air Quality Assessment Based on Human Olfactory Sensation. Journal of Architectural Planning and Environmental Engineering. 410: 9-19.

Janssen, J. E., T.J. Hill, J.E. Woods, and E.A.B. Maldonado. 1982. Ventilation for Control of Indoor Air Quality: A Case Study. Environment International. 8: 487-496.

Janu, G.J., J.D. Wenger, and C.G. Nesler. 1995. Strategies for Outdoor Airflow Control from a Systems Perspective. ASHRAE Transactions 101 (2): 631-643.

Ke, Y.-P. and S.A. Mumma. 1997. Using Carbon Dioxide Measurements to Determine Occupancy for Ventilation Controls. ASHRAE Transactions 103 (2): 365-374.

Klein S. 1994. TRNSYS, A Transient System Simulation Program, Version 14.1 Users Manual. University of Wisconsin-Madison: Solar Energy Laboratory.

Knoespel P, J. Mitchell, and W. Beckman. 1991. Macroscopic Model of Indoor Air Quality and Automatic Control of Ventilation Airflow. ASHRAE Transactions 97 (2): 1020-1030.

Kulmala, V., A. Salminen, G. Graeffe, K. Janka, J. Keskinen, and M. Rajala. 1984. Long-Term Monitoring of Indoor Air Quality and Controlled Ventilation in Public Buildings. Proceedings of the 3rd International Conference on Indoor Air Quality and Climate. Vol. 5: 435-441.

Kusuda, T. 1976. Control of Ventilation to Conserve Energy While Maintaining Acceptable Indoor Air Quality. ASHRAE Transactions 82 (1): 1169-1181.

Mansson, L. 1989. Annex 18 Demand Controlled Ventilating Systems. Proceedings of the 10th AIVC Conference Progress and Trends in Air Infiltration Research: 25-43

Mansson, L. 1994. Demand Controlled Ventilation Systems in Non-industrial Buildings. Proceedings of the European Conference on Energy Performance and Indoor Climate in Buildings.

Meckler, M. 1994. Demand-Control Ventilation Strategies for Acceptable IAQ. Heating/Piping/Air Conditioning. May: 71-74.

Meier, S. 1993. Mixed-Gas or CO_2 Sensors as a Reference Variable for Demand-Controlled Ventilation. Proceedings of the 6th International Conference on Indoor Air Quality and Climate. Vol. 5: 85-90.

Meier, S. 1995. Demand-Based Ventilation: Indoor Comfort at Lower Cost. Proceedings of Healthy Buildings ’95, 1743-1750.

Meier S. 1998. Demand-Controlled Ventilation – Requirements and Control Strategies. Proceedings of Roomvent ’98.

Nakahara, N. 1996. How to Design and Evaluate the HVAC for Better Indoor Air Quality with Sufficient Energy Saving. Proceedings of Indoor Air ‘96. Vol. 2: 3-15

NIOSH. 1987. A Guide to Safety in Confined Spaces. Cincinnati, OH: National Institute of Occupational Safety and Health, DHHS Publication No. 87-113.

Ogasawara, S., H. Taniguchi, and C. Sukehira. 1979. Effect of Energy Conservation by Controlled Ventilation: Case Study in a Department Store. Energy and Buildings 2: 3-8.

Okamoto, S., Muramatsu, S., Jeng, D.Y., Hadano, H., Nagawa, Y., and H. Futata. 1996. The Exploration of the Use of a Carbon Dioxide Sensor for Indoor Air Control. Proceedings of Indoor Air ‘96. Vol. 3: 333-338.

Persily, A.K. and W. S. Dols. 1990. The Relation of CO_2 Concentration to Office Building Ventilation. Air Change Rate and Airtightness in Buildings, ASTM STP 1067: 77-92.

Persily, A.K. 1993. Ventilation, Carbon Dioxide and ASHRAE Standard 62-1989. ASHRAE Journal. 35: 40-44.

Persily, A.K. 1997. Evaluating Building IAQ and Ventilation with Indoor Carbon Dioxide. ASHRAE Transactions. 103 (2): 193-204.

Potter, I.N. and W.B. Booth. 1994. CO_2 Controlled Mechanical Ventilated Systems. Technical Note. Bracknell, Berkshire: Building Services Research and Information Association, TN 12/94.

Raatschen, W., ed. 1990. Demand Controlled Ventilating System. State of the Art Review. Stockholm, Sweden: Swedish Council for Building Research, D9.

Rajhans, G.S. 1983. Indoor Air Quality and CO_2 Levels. Occupational Health in Ontario. 4: 160-167.

Rasmussen, C., G.H. Clausen, B. Berg-Munch and P.O. Fanger. 1985. The Influence of Human Activity on Ventilation Requirements for the Control of Body Odor. CLIMA 2000. Vol. 4: 357-362.

Reardon, J.T. and C.Y. Shaw. 1993. Carbon Dioxide Concentrations and Minimum Air Change Rates in a High-Rise Office Building. Indoor Environment 2: 337-343.

Reardon, J.T., C.Y. Shaw, and F. Vaculik. 1994. Air Change Rates and Carbon Dioxide Concentrations in a High-Rise Office Building. ASHRAE Transactions 100 (2): 1251-1263.

Ruud, S.H., P. Fahlen, and H. Andersson. 1991. Demand Controlled Ventilation - Full Scale Tests in a Conference Room. Proceedings of the 12th AIVC Conference Air Movement & Ventilation Control within Buildings: 187-200.

Schell, M.B., S.C. Turner, and R.O. Shim. 1998. Application of CO_2-Based Demand-Controlled Ventilation Using ASHRAE Standard 62: Optimizing Energy Use and Ventilation. ASHRAE Transactions 104: 1213-1225.

Schell, M. 2001. Proven Energy Savings with DCV retrofits. HPAC Engineering, February 2001.

Schell, M. and D. Int-Hout. 2001. Demand Control Ventilation Using CO_2. ASHRAE Journal. 43 (2): 18-29.

Schultz, K.J. and B.C. Krafthefer. 1993. CO_2-Based Ventilation Control: Choice of CO_2 Setpoint. ASHRAE Transactions 99 (1): 1548-1553.

Seppanen, O.A., W.J. Fisk and M. J. Mendell. 1999. Association of Ventilation Rates and CO_2 Concentrations with Health and Other Responses in Commercial and Institutional Buildings. Indoor Air. 9 (4): 226-252.

Shirey, D.B. and K. Rengarajan. 1996. Impacts of ASHRAE Standard 62-1989 on Small Florida Offices. ASHRAE Transactions 102 (1): 153-165.

Sodergren, D. 1982. A CO_2-Controlled Ventilation System. Environment International 8: 483-486.

Sorensen, B.R. 1996. Simulation of a Small VAV Plant. Proceedings of Indoor Air '96. Vol. 2: 199-204.

Strindehag, O., L. Norell, and S. Kvarnstrom. 1990. Improved Indoor Air Quality by Supervision of the CO_2 Content. Proceedings of the 5th International Conference on Indoor Air Quality and Climate. Vol. 4: 437-441.

Strindehag, O., and L. Norell. 1991. Long Term Experience with Demand-Controlled Ventilation Systems. Proceedings of ASHRAE IAQ 91, Healthy Buildings: 108-110.

Stymne, H., A. Mellin, and M. Sandberg. 1990. Dispersion Pattern of Carbon Dioxide from Human Sources - A Factor to Consider in Demand Controlled Ventilation Systems. Proceedings of the 5th International Conference on Indoor Air Quality and Climate. Vol. 4: 317-322.

Stymne, H., M. Sandberg, and M. Mattsson. 1991. Dispersion Pattern of Contaminants in a Displacement Ventilated Room - Implications for Demand Control. Proceedings of the 12th AIVC Conference Air Movement & Ventilation Control within Buildings: 173-189.

Telaire Systems, Inc. n.d. Application Guide for Carbon Dioxide (CO_2) Measurement & Control. Goleta, CA.

Turiel, I., C.D. Hollowell, and B.E. Thurston. 1979. Variable Ventilation Control Systems: Saving Energy and Maintaining Indoor Air Quality. Berkeley, CA: Lawrence Berkeley Laboratory. LBL- 9380.

Vaculik, F. and E.G. Plett. 1993. Carbon Dioxide Concentration-Based Ventilation Control. ASHRAE Transactions 99 (1): 1536-1547.

Walton, G. 1989. AIRNET - A Computer Program for Building Airflow Network Modeling. Gaithersburg, MD: National Institute of Standards and Technology, NISTIR 89-4072.

Wang S. and X. Jin. 1998. CO_2-Based Occupancy Detection for On-Line Outdoor Air Flow Control. Indoor & Built Environment; 7:165-181.

Wang S. and X. Jin. 1999. Experimental Validation of CO_2-Based Occupancy Detection for Demand-Controlled Ventilation. Indoor & Built Environment; 8:377-391.

Warren, B.F. 1982. Energy Savings in Buildings by Control of Ventilation as a Function of Indoor Carbon Dioxide Concentration. Building Services Engineering Research & Technology 3 (1): 4-12.

Warren, B.F. and N.C. Harper. 1991. Demand Controlled Ventilation by Room CO_2 Concentration: A Comparison of Simulated Energy Savings in an Auditorium Space. Energy and Buildings 17: 87-96.

Woods, J.E., G. Winakor, E.A.B. Maldonado, and S. Kipp. 1982. Subjective and Objective Evaluation of a CO_2-Controlled Variable Ventilation System. ASHRAE Transactions 88 (1): 1385-1393.

Wright, G. 1997. DCV expands options for improving IAQ. Building Design & Construction, January 1997.

Zamboni, M., O. Berchtold, C. Filleux, J. Fehlmann, and F. Drangsholt. 1991. Demand Controlled Ventilation - An Application to Auditoria. Proceedings of the 12th AIVC Conference Air Movement & Ventilation Control within Buildings: 143-155.

Zmeureanu, R., and F. Haghighat. 1995. Computer Simulation of the Impact of CO_2-based Ventilation Control. Proceeding of Building Simulation '95: 206-212.

Appendix A: CEC RFP Issues

The California Energy Commission (CEC) Public Interest Energy Research (PIER) Request for Proposals for the Buildings Energy Efficiency Program Area identified four key issues of concern. These four issues identify energy problems facing buildings in California and present opportunities to have a significant positive impact. This appendix discusses the relationship of the application of CO_2-based DCV systems to the four key issues based on information in this report.

Issue #1 Energy consumption is rapidly increasing in hotter, inland areas as new building construction increases in these areas.

A key intent of CO_2-based DCV systems is the reduction of energy consumed to cool and heat ventilation air in buildings. As discussed in the review of case studies in this report, they are capable of achieving such reductions for many building types in a variety of climates. Fortuitously, DCV systems may be very well suited to reducing energy consumption in the hotter, inland areas of California as evidenced by a recent simulation study (Brandemuehl and Braun 1999).

Brandemuehl and Braun simulated the energy impacts of DCV systems and economizers for four building types (office, retail, school, and restaurant) in a variety of locations including Sacramento. They reported potential electrical energy savings of 17 % for the restaurant in Sacramento with both a DCV system and an economizer. Equal savings were due to the DCV system and the economizer. For the other building types in Sacramento, the combined DCV and economizer systems reduced electrical energy consumption by 10 % for the office, 19 % for the retail, and 18 % for the school. Since Brandemuehl and Braun and others have reported larger potential savings for humid locations and hotter locations, a DCV system with an economizer is likely to save even more electrical energy in locations such as Fresno and Palm Springs, climates combining warm weather and high humidity levels. The use of carefully scheduled purge ventilation with DCV could also increase energy savings by reducing ventilation during the hottest times of the day.

Issue #2 Development of energy efficient products and services needs to adequately consider non-energy benefits, such as comfort, productivity, durability, and decreased maintenance.

Since CO_2-based DCV systems directly affect building ventilation rates, the potential exists to have a significant impact on occupant comfort and productivity. That impact could be either positive or negative depending on the DCV system design, installation, operation and maintenance. CO_2-based DCV systems can indeed have a positive impact on IAQ that is not always considered, when building zones are occupied by more people than the number on which the ventilation system design is based. At such times, a DCV system will result in improved IAQ by providing an appropriate amount of ventilation to the space. Additionally, ventilation systems may operate with lower ventilation effectiveness than the design criteria. Again, a DCV system can increase ventilation rates in such situations. While it is not possible to estimate potential impacts on productivity for any given building, Fisk and Rosenfeld (1997) have estimated that nationwide impacts of better indoor environments are in the billions of dollars.

Since DCV systems adjust ventilation rates based on measured concentrations of CO_2 generated by building occupants, they do not directly guarantee satisfactory indoor air quality (IAQ) due to the presence of non-occupant generated contaminants. This issue results in a concern by some that DCV could result in poor IAQ, which could negatively impact comfort and productivity. Certain steps need to be taken to avoid the occurrence of such a negative impact. The most fundamental step is to implement the same good IAQ practices that should be applied to all commercial buildings. These practices include reduction of contaminant sources, proper installation and maintenance of equipment, etc. Additional steps that should be taken for DCV systems include appropriate selection of control algorithms and setpoints, thoughtful consideration of expected contaminant sources, establishment of minimum base and/or purge ventilation rates and schedules, system commissioning, and proper maintenance and calibration of CO_2 sensors.

Issue #3 Building design, construction, and operation of energy-related features can affect public health and safety.

The above discussion addressing Issue #2 also applies to public health. CO_2-based DCV systems could have either a negative or positive impact on public health, and therefore care needs to be taken in their application. In addition, DCV can have a very positive impact in lessening the moisture load in non-residential buildings in humid climates. Since most of the moisture load for many non-residential buildings is brought into a building through ventilation, reducing excess ventilation during times of reduced building occupancy can reduce this moisture load. This reduction in moisture load can save energy and money by eliminating the need for special equipment.

Issue #4 Investments in energy efficiency can affect building and housing affordability and value, and the state's economy.

As discussed in response to Issue #1, CO_2-based DCV systems can reduce building heating and cooling energy use and, therefore, reduce operating costs to improve building affordability and value. However, these potential savings will vary widely depending on building type, climate, occupancy density and patterns, other HVAC system characteristics, and other factors. While knowledge of these important parameters is growing, more work is needed to identify the best opportunities for energy savings. No significant impacts are expected on the energy-related costs of construction.

References

Brandemuehl, M.J. and J.E. Braun. The Impact of Demand-Controlled and Economizer Ventilation Strategies on Energy Use in Buildings (1999) ASHRAE Transactions 105 (2): 39-50.

Fisk, W.J. and A.H. Rosenfeld. Estimates of Improved Productivity and Health from Better Indoor Environments (1997) Indoor Air 1997; 7:158-172.

Appendix B: Preliminary Application Guidance

The intent of this section is to summarize the currently available guidance on the application of CO_2-based DCV while acknowledging that significant knowledge gaps still remain. This summary is based on guidance in various publications including lessons learned from published case studies. Future phases of this study are intended to fill in many of these knowledge gaps and to develop more definitive guidance. Furthermore, it is not the intent of this section to describe a system that meets ASHRAE Standard 62 or any other ventilation standard.

Target buildings

The intent of a CO_2-based DCV is to save cooling, heating, and fan energy use compared to ventilation at a constant rate based on design occupancy, while assuring adequate ventilation rates for IAQ control. While CO_2-based DCV systems are likely to save at least some energy for nearly all buildings and climates, the amount of energy saved can vary dramatically depending on the climate, occupancy, operating hours, and other building and HVAC system features. The greatest energy savings are likely to occur in buildings with large heating loads or large cooling loads that have dense occupancies that vary unpredictably. This section highlights circumstances, specifically buildings and spaces, where CO_2 DCV appears to make the most sense.

Good candidates

Building or spaces where the occupants are likely to be the only significant source of CO_2.

Buildings or spaces with dense, unpredictably variable occupancies.

Buildings or spaces in climates that have significant heating or cooling loads.

Variably occupied spaces that have independent outdoor air supply capability such as a conference rooms within buildings in which the building as a whole may not be a good candidate.

Poor candidates

Buildings or spaces where ventilation requirements are dominated by non-occupant generated pollutants.

Buildings or spaces with significant sources of CO_2 other than occupants. Using CO_2 as the control variable in such applications will not necessarily result in unacceptable IAQ but rather could cause excessive ventilation rates. In such cases, it may be possible to control the ventilation rate based on another measured parameter.

Buildings or spaces with any CO_2 removal mechanisms other than ventilation, such as air cleaning of CO_2.

Remaining Issues/Questions

Is it economical for CO_2-based DCV to be applied in mild climates even with an economizer?

While CO_2-based DCV may not be cost-effective for most buildings with no heating demand in low humidity climates, CO_2-based DCV may still be considered for potential IAQ benefits or ensuring ‘adequate’ ventilation. How significant are these IAQ benefits?

CO_2 DCV Technology

Most CO_2 sensors used in DCV systems today are based on non-dispersive infrared (NDIR) or photometric detection. Both technologies can be affected by light source aging. The former approach may be sensitive to particle buildup on the sensor while the latter could be affected by vibration or atmospheric pressure changes.

Good sensor attributes and application

Appropriate measurement ranges for ventilation control.

Calibrated according to manufacturer recommendations. An automated calibration system that uses overnight baseline CO_2 readings may be considered.

Located in occupied zones when an appropriate location is available..

Poor sensor attributes and application

Should not use CO_2 monitors that are not intended for control systems.

Should not be located near doors, windows, air intakes or exhausts, or in close proximity to occupants.

A single sensor located in a common return should not be used to control ventilation rates for spaces with very different expected occupancies.

Remaining Issues/Questions

Is it acceptable to use a single sensor in a common return to control the ventilation rate for multiple zones with similar expected occupancies? Can a lower setpoint compensate for differences in concentrations between zones? How much could this approach reduce energy savings?

Are there significant advantages to using a single sensor with multiple measurement locations compared to multiple sensors?

<u>Control Algorithms</u>

Control strategies for CO_2-based DCV include simple setpoint control where the ventilation rate is increased or decreased depending on the indoor CO_2 concentration, proportional control in which the ventilation rate is proportional to the CO_2 concentration, PI (proportional-integral) or PID (proportional-integral-derivative) control which can adjust more quickly to changes in the CO_2 concentration, and algorithms that aim to maintain a constant ventilation rate per person at all times.

Good practice

Control strategies should be chosen based on the expected occupancy patterns. Control algorithms that can adjust ventilation rates more quickly should be considered for spaces with low density occupancy or where changes in occupancy may be gradual .

Remaining Issues/Questions

Should CO_2 setpoints be varied for buildings with occupants whose CO_2 generation is expected to vary significantly from that of adults doing office work? It is necessary to account for the lower CO_2 generation of children in schools or the higher generation of very active adults? Is a control algorithm that maintains a constant ventilation rate per person necessary for acceptable IAQ?

<u>Other Contaminants</u>

CO_2-based DCV systems should include a strategy to provide for sufficient ventilation, or other means, to control concentrations of non-occupant generated contaminants. Ideally, an analysis of non-occupant sources would indicate the appropriate ventilation and other IAQ control technologies needed to maintain the resulting concentrations of contaminants within acceptable limits. However, the information needed to perform such an analysis will not likely be available in most situations.

Good practice

CO_2-based DCV should include a strategy of ventilation and other IAQ control technologies to control non-occupant generated contaminants. It may be possible to control known pollutant sources through local ventilation or air cleaning.

Poor practice

CO_2-based DCV may not be appropriate in spaces where smoking is permitted.

Remaining Issues/Questions

What level of minimum ventilation, if any, is needed? Although recommendations have been made to maintain a minimum ventilation rate of 10 % to 50 % of the design rate, no general rule of thumb is available. Minimum ventilation rates should be established based on expected types and strengths of pollutant sources, and other IAQ control technologies employed.

Can scheduled purges replace the minimum ventilation rate? It may be possible to maintain average contaminant concentrations below the same limits that would result from a constant ventilation rate by scheduling purges at appropriate times, such as prior to occupancy. Such a strategy may save energy if the purges can be scheduled in the afternoon during heating season and during the morning in the cooling season.

Other Considerations

The selection and design of a CO_2-based DCV system cannot be viewed in isolation. The air quality and energy performance of a DCV system will impact and be impacted by other building and HVAC systems. While no comprehensive listing of potential interactions is available, significant interactions can occur with economizers, displacement ventilation, and other technologies.

Good practice

In buildings with an economizer cycle, the economizer should be allowed to override the DCV system at times when the additional ventilation would provide 'free' cooling.

Consider installation of an outdoor CO_2 sensor if outdoor levels are expected to deviate significantly (more than about 20 %) from 720 mg/m^3 (400 ppm(v)). The outdoor CO_2 concentrations can be assumed to be 720 mg/m^3 (400 ppm(v)) for most applications, but urban areas may have local effects resulting in higher levels. The higher outdoor level could result in overventilation and it may be economical to install an additional sensor to control the difference between indoor and outdoor concentration directly. Such an installation could also be required by applicable standards or codes.

"Poor" practice

CO_2-based DCV may not be appropriate in buildings in mild climates (little or no heating demand and low humidity) unless an economizer is also used.

Remaining Issues/Questions

If a displacement ventilation system is used, where should the sensor be located and can the setpoint be lowered?

VSAT – Ventilation Strategy Assessment Tool

Submitted to

California Energy Commission

As Deliverables 3.1.2, 3.2.1, and 4.2.2

Prepared by

James E. Braun and Kevin Mercer
Purdue University

Revised August 2003

Table of Contents

SECTION 1: INTRODUCTION

This report describes a simulation tool (VSAT – Ventilation Strategy Assessment Tool) that estimates cost savings associated with different ventilation strategies for small commercial buildings. A set of prototypical buildings and equipment is part of the model. The tool is not meant for design or retrofit analysis of a specific building. It does provide a quick assessment of alternative ventilation technologies for common building types and specific locations with minimal input requirements.

Figure 1 shows a schematic of a small commercial building and HVAC system. The buildings currently considered within VSAT include a small office building, a sit-down restaurant, a retail store, a school class wing, a school auditorium, a school gymnasium, and a school library. All of these buildings are considered to be single zone with a slab on grade (no basement or crawl space). VSAT considers only packaged HVAC equipment, such as rooftop air conditioners with integrated cooling equipment, heating equipment, supply fan, and ventilation. Modifications to the ventilation system are the focus of the tool's evaluation. A basic ventilation system (shown within the box of Figure 1) consists of ambient supply, exhaust, and return ducts and dampers. The different ventilation strategies that are considered by VSAT are: 1) fixed ventilation rates with no economizer, 2) fixed ventilation rates with a differential enthalpy economizer, 3) demand-controlled ventilation with an economizer, 4) fixed ventilation rates with heat recovery using an enthalpy exchanger, 5) fixed ventilation rates with heat recovery using a heat pump, 6) night ventilation precooling, 7) night ventilation precooling with an economizer, and 8) night ventilation precooling with demand-control ventilation and an economizer. Details about these strategies are given in later sections.

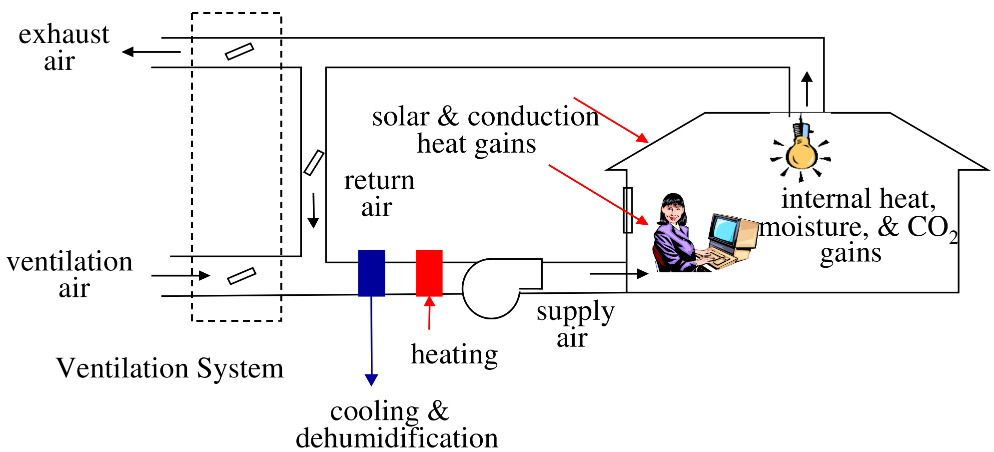


Figure 1. Schematic of a Small Commercial Building and HVAC System

VSAT is derived from a simulation tool that was developed by Braun and Brandemuehl (2002) called the Savings Estimator. It performs calculations for each hour of the year using fairly detailed models and TMY2 or California Climate Zone weather data. The goal in developing VSAT was to have a fast, robust simulation tool for comparison of ventilation options that could consider large parametric studies involving different systems and locations. Existing commercial simulation tools do not consider all of the ventilation options of interest for this project.

Figure 2 shows an approximate flow diagram for the modeling approach used within VSAT. Given a physical building description, an occupancy schedule, and thermostat control strategy, the building model provides hourly estimates of the sensible cooling and heating requirements needed to keep the zone temperatures at cooling and heating setpoints. It involves calculation of transient heat transfer from the building structure and internal sources (e.g., lights, people, and equipment). The air distribution model solves energy and mass balances for the zone and air distribution system and determines mixed air conditions supplied to the equipment. The mixed air condition supplied to the primary HVAC equipment depends upon the ventilation strategy employed. The zone temperatures are outputs from the building model, whereas the zone and return air humidities and CO_2 concentrations are calculated by the air distribution model. The equipment model uses entering conditions and the sensible cooling requirement to determine the average supply air conditions. The entering and exit air conditions for the air distribution and equipment models are determined iteratively at each timestep of the simulation using a non-linear equation solver. Details of each of the component models are described in later sections.

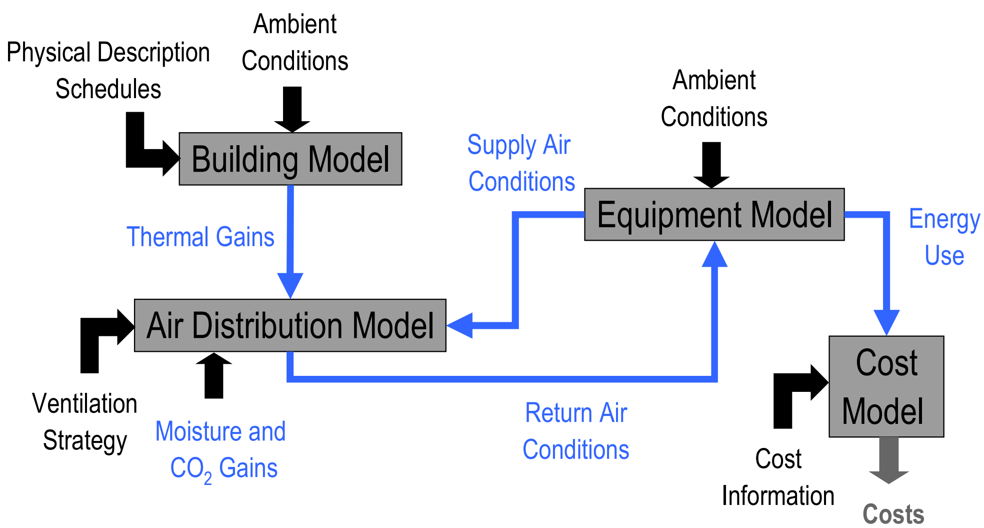


Figure 2. Schematic of VSAT Modeling Approach

SECTION 2: BUILDING MODEL

The space loads are based on the building physical characteristics, operating schedule, occupancy patterns, and space setpoints. The total sensible loads are calculated from an energy balance on the zone air for a given temperature setpoint with individual heat gains from walls, roof, floor, windows, internal gains, and infiltration. The following sections describe individual models for each of these elements and the overall strategy for estimating sensible cooling and heating requirements for the building.

2.1 Model Description

2.1.1 Exterior Walls and Roofs

Figure 3 shows the heat transfer rates and nomenclature associated with an external wall or roof (j[th] wall). One-dimensional heat transfer is assumed. The symbols $\dot{Q}$ and T denote heat transfer rates and temperatures, respectively. The subscripts i and o refer to conditions at the inside and outside of the wall, respectively. The subscript c refers to convection, whereas r denotes radiation. The subscript s refers to conduction within the wall at the surface (inside or outside).

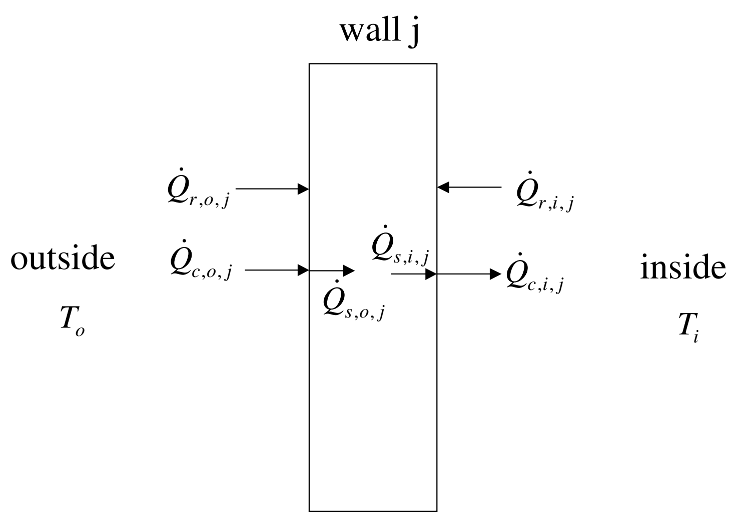


Figure 3. Heat transfer rates for an external wall

Radiation at the outside of the wall is due to solar (short-wave radiation) and long-wave radiation exchange with the sky and other surfaces. Long-wave radiation is assumed to occur between the wall surface and other surfaces that are at the ambient temperature (T_o). Furthermore, the radiation is linearized so that a radiation heat transfer coefficient is determined at a representative mean temperature. The long-wave radiation is combined with the convection using a combined convection and long-wave radiation heat transfer coefficient. With these assumptions, the effective outside convection (convection and long-wave radiation) and radiation (short-wave only) for wall j are calculated as

$$\dot{Q}_{c,o,j} = h_o A_j (T_o - T_{s,o,j}) \tag{2.1}$$

$$\dot{Q}_{r,o,j} = \alpha_o A_j I_{o,j} \tag{2.2}$$

where h_o is the outside heat transfer coefficient (convection and long-wave radiation), A is wall surface area, α_o is the absorptance for solar radiation of the outside surface, I_o is the instantaneous radiation incident upon the outside surface. The outside heat transfer coefficient and absorptance are assumed to be constant, independent of operating conditions (e.g., wind speed).

The conduction at the outside surface of the wall is equal to the sum of the convective and radiative gains. In order to simplify the transient heat transfer calculations, an equivalent outside air temperature is defined that would give the correct heat transfer rate in the absence of the solar radiation gains. This is commonly referred to as the sol-air temperature and is calculated as

$$T_{eq,o,j} = T_o + \frac{\alpha_o I_{o,j}}{h_o} \tag{2.3}$$

With this definition, the conduction heat transfer rate at the outside surface is

$$\dot{Q}_{s,o,j} = h_o A_j (T_{eq,o,j} - T_{s,o,j}) \tag{2.4}$$

A similar approach is followed for the inside surface: long-wave radiation is assumed to occur between each wall surface and other wall surfaces that are at the inside air temperature (T_i); long-wave radiation exchange with other surfaces is linearized so that a radiation heat transfer coefficient is determined at a representative mean temperature; long-wave radiation is combined with convection using a combined convection and long-wave radiation heat transfer coefficient; an equivalent inside air temperature is defined that would give the correct heat transfer rate in the absence of the internal radiation gains (from solar through windows and internal sources). With these assumptions, the conduction heat transfer rate at the inside wall surface is

$$\dot{Q}_{s,i,j} = h_i A_j (T_{eq,i,j} - T_{s,i,j}) \tag{2.5}$$

where

$$T_{eq,i,j} = T_i + \frac{\dot{q}_{g,r}}{h_i} \tag{2.6}$$

and where h_i is the inside heat transfer coefficient (convection and long-wave radiation) and $\dot{q}_{g,r}$ is the absorbed radiation flux due to internal sources and solar radiation transmitted through windows.

The transient heat transfer problem for a wall can be represented using an electrical analog. Figure 4 shows a simple two-node representation (two state variables) for a wall subjected to time-varying temperature boundary conditions. Outside and inside radiation gains are handled with an equivalent air temperature. In this representation, R represents a thermal resistance and C is a thermal capacitance. The total thermal resistance ($R_1 + R_2 + R_3$) includes the thermal resistance between the outside air and the wall (combined convection and long-wave radiation), the conduction resistance within the wall and the thermal convection resistance between the wall and the building interior. The capacitors incorporate the total capacitance of the wall material. For this simple representation, the physical location of the nodes has a significant effect on the model predictions. Chaturvedi and Braun (2002) found that 2 or 3 nodes were sufficient to provide accurate transient predictions if the location of the nodes were optimized. For best results, the outside and inside resistances should include the air resistance and a portion of the material within the wall.

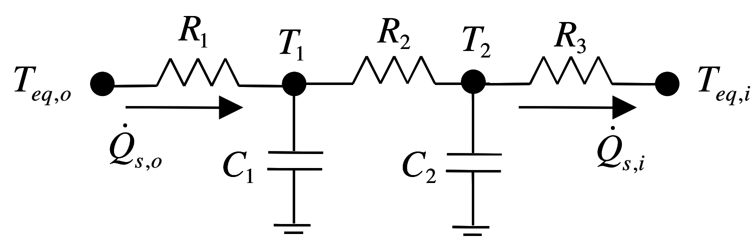


Figure 4. Thermal network representation of an external wall

The electrical circuit can easily be represented in state-space form as

$$\frac{d\vec{x}}{d\tau} = \hat{A}\vec{x} + \hat{B}\vec{u} \tag{2.7}$$

$$y = \vec{c}^{\,T}\vec{x} + \vec{d}^{\,T}\vec{u} \tag{2.8}$$

where $\vec{x}$ = vector of state variables
$\vec{u}$ = vector of inputs
y = output variable
$\hat{A}$ = constant coefficient matrix
$\hat{B}$ = constant coefficient matrix
$\vec{c}$ = constant coefficient vector
$\vec{d}$ = constant coefficient vector
τ = time

For a wall, the desired output variable is the rate of conduction heat transfer at the inside surface ($\dot{Q}_{s,i}$). The state vector contains temperatures of "nodes" within the structure of the wall, the input vector consists of the equivalent inside and outside air temperatures ($T_{eq,i}$ and $T_{eq,o}$), and coefficient matrices and vectors contain the physical characteristics of the wall (i.e., the R's and C's).

The state-space formulation could be solved at each timestep of a simulation. However, the computation can be significantly reduced if the state-space formulation is converted to a transfer function representation. Seem et al. (1989) presented a technique for determining an equivalent transfer function representation from the state-space representation that involves the exact solution to the set of first-order differential equations with the inputs modeled as continuous, piecewise linear functions. This approach is used within VSAT for a one-hour timestep to determine a transfer function equation at the beginning of the simulation. After the transfer function has been developed, then the solution for the output at any time t is of the form

$$y(t) = \sum_{k=0}^{N_{state}} \vec{S}_k^T \cdot \vec{u}_{t-k\Delta\tau} - \sum_{k=1}^{N_{state}} e_k \cdot y(t - k\Delta\tau) \tag{2.9}$$

where N_{state} = number of state variables

$\vec{S}_k$ = vector containing transfer function coefficients for the input vector k timesteps prior to the current time t

e_k = transfer function coefficient for the zone sensible load for k timesteps prior to the current time t

$\Delta\tau$ = time step (one hour for VSAT)

At the beginning of the simulation, the vectors $\vec{S}_k$ for k = 0 to N_{state} are determined as

$$\begin{aligned} \vec{S}_0 &= \vec{c}\hat{R}_0\Gamma_2 + \vec{d} \\ \vec{S}_j &= \vec{c}\left[\hat{R}_{j-1}(\Gamma_1 - \Gamma_2) + \hat{R}_j\Gamma_2\right] + e_j\vec{d} \quad \text{for } 1 \le j \le (N_{state} - 1) \\ \vec{S}_{N_{state}} &= \vec{c}\hat{R}_{N_{state}-1}(\Gamma_1 - \Gamma_2) + e_{N_{state}}\vec{d} \end{aligned} \tag{2.10}$$

where

$$\begin{aligned} \Gamma_1 &= \hat{A}^{-1}\left(\Phi - \hat{I}\right)\hat{B} \\ \Gamma_2 &= \hat{A}^{-1}\left[\frac{\Gamma_1}{\Delta\tau} - \hat{B}\right] \end{aligned} \tag{2.11}$$

where $\hat{I}$ is the identity matrix, $\Delta\tau$ is the simulation time step (one hour for this study), and

$$\begin{aligned} \Phi &= e^{\hat{A}\Delta\tau} \\ e^{\hat{A}\Delta\tau} &= \hat{I} + \hat{A}\Delta\tau + \frac{\hat{A}^2(\Delta\tau)^2}{2!} + \frac{\hat{A}^3(\Delta\tau)^3}{3!} + \ldots + \frac{\hat{A}^n(\Delta\tau)^n}{n!} + \ldots \end{aligned} \tag{2.12}$$

Seem et al. (1989) presented an efficient algorithm for evaluating $e^{\hat{A}\Delta\tau}$ in equation 2.12 that is used within VSAT. The matrices $\hat{R}_j$ used in the determination of $\vec{S}_k$ and the e_j transfer function coefficients are determined recursively as

$$
\begin{aligned}
&\hat{R}_0 = \hat{I} && e_1 = -\frac{Tr\left(\Phi\hat{R}_0\right)}{1} \\
&\hat{R}_1 = \Phi\hat{R}_0 + e_1\hat{I} && e_2 = -\frac{Tr\left(\Phi\hat{R}_1\right)}{2} \\
&\hat{R}_2 = \Phi\hat{R}_1 + e_2\hat{I} && e_3 = -\frac{Tr\left(\Phi\hat{R}_2\right)}{3} \\
&\vdots && \vdots \\
&\hat{R}_{N_{state}-1} = \Phi\hat{R}_{N_{state}-2} + e_{N_{state}-1}\hat{I} && e_{N_{state}} = -\frac{Tr\left(\Phi\hat{R}_{N_{state}-1}\right)}{N_{state}}
\end{aligned}
\tag{2.13}
$$

where Tr() is the trace of the matrix (the sum of the diagonal elements).

The transfer function representation gives the wall conduction at the inside surface for any wall j. The heat transfer to the inside air due to wall j is then

$$\dot{Q}_{i,j} = \dot{Q}_{s,i,j} + A_j\dot{q}_{g,r} \tag{2.14}$$

2.1.2 Floor Slabs

Slab on grade floors are modeled using a similar formulation as for exterior walls. However, the exterior of the floor is exposed to the ground so that there is no convection, solar radiation, or long-wave radiation. Furthermore, the predominant mechanism for heat loss or gain is heat transfer at the perimeter of the slab. The transfer function of equation 2.9 is used to determine the conduction heat transfer at the inside surface for floors. However, the bottom side of the floor is assumed to be adiabatic (infinite resistance for heat transfer between the outside floor surface and the ground). The primary mode for heat transfer to and from the ambient is through the perimeter of the slab. Perimeter heat transfer is assumed to be quasi-steady state from the ambient to the inside air across a resistance that is based upon the slab perimeter heat loss factor (ASHRAE, 2001). The combined heat transfer to the inside air from the floor is then

$$\dot{Q}_i = \dot{Q}_{s,i} + A\dot{q}_{g,r} + F_p \cdot P \cdot (T_o - T_i) \tag{2.15}$$

where F_p is the slab perimeter heat loss factor and P is the perimeter of the slab.

2.1.3 Interior Walls

An interior wall differs from an exterior wall in that the inside boundary conditions are experienced on both sides of the wall. The transfer function of equation 2.9 is used to determine the conduction heat transfer at the inside surfaces for interior walls with both boundary conditions given by equation 2.6. Interior walls are assumed to be symmetric with identical boundary conditions, so that the total heat transfer to the air from both surfaces is

$$\dot{Q}_i = 2 \cdot \left(\dot{Q}_{s,i} + A\dot{q}_{g,r}\right) \tag{2.16}$$

where A is the surface area for one face and $\dot{Q}_{s,i}$ is the conduction heat transfer rate for one surface of the wall.

Interior walls/furnishings are represented with a single node (capacitance) having a total surface area equal to twice the total floor area, a mass of 25 lbm/ft^2, and an average specific heat of 0.2 Btu/lbm-F.

2.1.4 Windows

Figure 5 shows the relevant heat transfer rates for the kth window. Windows are considered as quasi-steady-state elements that provide heat gains due to both solar transmission and conduction. Similar to walls, long-wave radiation is combined with convection using combined heat transfer coefficients at the inside and outside surfaces. Solar radiation passing through the window is partially absorbed and mostly transmitted. The overall absorptance and transmittance for solar radiation of the window are α and τ, respectively.

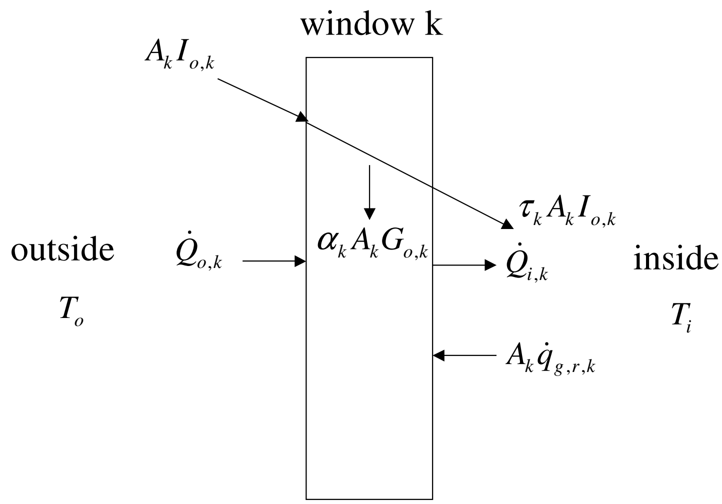


Figure 5. Heat transfer rates for a window

Assuming that the absorption of solar radiation occurs at the outside surface and absorption of internal radiative gains occurs at the inside surface, then the heat transfer rate by conduction through the glass is

$$\dot{Q}_{s,i,k} = U_k A_k (T_{eq,o,k} - T_{eq,o,k}) \tag{2.17}$$

where U is the overall unit conductance for the window. The equivalent inside and outside air temperatures ($T_{eq,i}$ and $T_{eq,o}$) are evaluated using equations 2.3 and 2.6, respectively. Then, the total heat gains through the window are

$$\dot{Q}_{win,k} = U_k A_k (T_{eq,o,k} - T_{eq,i,k}) + \tau_k \cdot A_k \cdot I_{o,k} + A_k \cdot \dot{q}_{g,r,k} \tag{2.18}$$

It is more common to have data for window shading coefficients than for window transmittances. The shading coefficient accounts for both solar transmission and solar absorption. In this formulation, the total heat gain to the air due to the window is given as

$$\dot{Q}_{win,k} = U_k A_k (T_o - T_{eq,i,k}) + SHGC_k \cdot A_k \cdot I_{o,k} + A_k \cdot \dot{q}_{g,r,k} \tag{2.19}$$

where *SHGC* is the solar heat gain coefficient defined as

$$SHGC = \tau + \frac{U\alpha}{h_o} \tag{2.20}$$

where h_o is the outside heat transfer coefficient (combined convection and long-wave radiation). Equations 2.18 and 2.19 are equivalent.

The shading coefficient is defined as

$$SC = \frac{SHGC}{SHGC_{ref}} \tag{2.21}$$

where $SHGC_{ref}$ is the solar heat gain coefficient for a single pane of double strength glass, which has a value of 0.87. In general, the shading coefficient can account for multiple glazings, different types of glazing materials, and indoor shading devices.

Using the definition of shading coefficient, equation 2.19 can be rewritten as

$$\dot{Q}_{win,k} = U_k A_k (T_o - T_{eq,i,k}) + SC \cdot SHGC_{ref} \cdot A_k \cdot I_{o,k} + A_k \cdot \dot{q}_{g,r,k} \tag{2.22}$$

The concept of a shading coefficient was developed for building models where the heat gains due to solar radiation are added directly to the air. In reality, solar transmission through windows leads to solar absorptance on other interior surfaces, whereas solar absorption in windows leads to increased convection to the air by the window. Although it is not strictly correct, VSAT uses the total solar gains determined with a shading coefficient and distributes them to other internal surfaces. With this approach, the window solar transmission and convection to the air are determined as

$$\dot{Q}_{t,k} = SC \cdot SHGC_{ref} \cdot A_k \cdot I_{o,k} \tag{2.23}$$

$$\dot{Q}_{i,k} = U_k A_k (T_o - T_{eq,i,k}) + A_k \cdot \dot{q}_{g,r,k} \tag{2.24}$$

VSAT assumes constant values for the shading coefficient and overall window unit conductance. Solar transmission through windows is distributed solely to the floor with a uniform heat flux.

2.1.5 Infiltration

Infiltration is a relatively small effect for commercial buildings and is modeled with a constant flow rate that is based upon a specified volumetric flow rate per unit floor area. The default value is 0.05 cfm/ft^2, but can be changed. For a building with 10-foot ceiling height, this infiltration rate corresponds to 0.3 air changes per hour.

The sensible and latent heat gains due to infiltration are determined as

$$\dot{Q}_{\inf,s} = \dot{m}_{\inf} C_{pm} (T_o - T_i) \tag{2.25}$$

$$\dot{Q}_{\inf,L} = \dot{m}_{\inf} h_{fg} (\omega_o - \omega_i) \tag{2.26}$$

where C_{pm} is the moist air specific heat, h_{fg} is the heat of vaporization of water, ω_o is the humidity ratio of the outside air, and ω_i is the humidity ratio of the inside air.

2.1.7 Internal Gains

Internal gains due to lights, equipment, and people vary according to an occupancy schedule that is specified. The specific values of the heat gains and the proportion of gains from people that influence latent loads vary according to building type (see Prototypical Building Descriptions). For people and lights, 50% of the heat gains are assumed to be radiative and 50% convective. All the gains from equipment (e.g, computers) are assumed to be convective. The radiative internal gains are distributed with an even heat flux to all internal surfaces (including windows).

2.1.8 Zone Loads

At any time, the sensible cooling (+) or heating (-) required to keep the zone temperature at a specified setpoint is determined as

$$\dot{Q}_z = \sum_{j=1}^{walls} \dot{Q}_{i,j} + \sum_{k=1}^{windows} \dot{Q}_{i,k} + \dot{Q}_{g,c} + \dot{Q}_{s,\inf} \tag{2.27}$$

where $\dot{Q}_{g,c}$ is the total convective heat gain due to lights, people, and equipment.

Separate temperature setpoints are specified for heating and cooling and the temperature can float in between with no required cooling or heating. In order to evaluate whether heating or cooling is required for a given time step, it is necessary to determine the zone temperature where the sensible cooling requirement for the equipment is equal to zero. In the absence of ventilation (unoccupied mode) then equation 2.27 would be solved inversely for the floating inside air temperature with $\dot{Q}_z$ set equal to zero. If the calculated zone temperature is less than the heating setpoint, then heating is required and equation 2.27 is evaluated using the heating setpoint. If the calculated zone temperature is greater than the cooling setpoint, then cooling is required and equation 2.27 is evaluated using the cooling setpoint. If the calculated temperature is between the setpoints, then the zone temperature is floating and the zone sensible cooling and heating requirement are zero. The case where the fans operate continuously with a ventilation load (unoccupied mode) is considered in section 4.

When there is a sensible cooling requirement, then the cooling equipment also provides latent cooling and it is necessary to know the latent loads for the zone. In this case, the zone latent gains are the sum of the latent gains due to people and due to infiltration.

2.1.9 Solar Radiation Processing

The weather data files used by VSAT contain hourly values of global horizontal radiation and direct normal radiation. The horizontal radiation is used for the roof, but it is necessary to calculate incident radiation on vertical surfaces for external walls. The total incident radiation for vertical surfaces is determined as

$$I_o = I_{DN} \sin(\theta_z) \cdot \cos(\gamma_s - \gamma) + \frac{I_D}{2} + \rho_g I_H \tag{2.28}$$

where I_{DN} is beam radiation that is measured normal to the line of sight to the sun, θ_z is the zenith angle, γ_s is the solar azimuth angle, γ is the surface azimuth angle, I_D is sky diffuse radiation, ρ_g is ground reflectance, and I_H is total radiation incident upon a horizontal surface. Zenith is the angle between the vertical and the line of site to the sun. Solar azimuth is the angle between the local meridian and the projection of the line of sight to the sun onto the horizontal plane. Zero solar azimuth is facing the equator, west is positive, while east is negative. The zenith and solar azimuth angle are calculated using relationships given in Duffie and Beckman (1980). The surface azimuth is the angle between the local meridian and the projection of the normal to the surface onto the horizontal plane (0 for south facing, -90 for east facing +90 for west facing, and +180 for north facing). The ground reflectance is assumed to have a constant value of 0.2, which is representative of summer conditions. The sky diffuse radiation is calculated from the

$$I_D = I_H - I_{DN} \cos(\theta_z) \tag{2.29}$$

2.2 Prototypical Building Descriptions

Seven different types of buildings are considered in VSAT: small office, school class wing, retail store, restaurant dining area, school gymnasium, school library, and school auditorium. Descriptions for these buildings were obtained from prototypical building descriptions of commercial building prototypes developed by Lawrence Berkeley National Laboratory (Huang, et al. 1990 & Huang, et al. 1995). These reports served as the primary sources for prototypical building data. However, additional information was obtained from DOE-2 input files used by the researchers for their studies.

Tables 1 - 7 contain information on the geometry, construction materials, and internal gains used in modeling the different buildings. Although not given in these tables, the walls, roofs and floors include inside air and outside air thermal resistances. The window R-value includes the effects of the window construction and inside and outside air resistances. Table 8 lists the properties of all construction materials and the air resistances. The geometry of each of the buildings is assumed to be rectangular with four sides and is specified with the following parameters: 1) floor area, 2) number of stories, 3) aspect ratio, 4) ratio of exterior perimeter to total perimeter, 5) wall height and 6) ratio of glass area to wall area. The aspect ratio is the ratio of the width to the length of the building. However, exterior perimeter and glass areas are assumed to be equally distributed on all sides of the building, giving equal exposure of exterior walls and windows to incident solar radiation. The four exterior walls face north, south, east, and west.

The user can specify occupancy schedules, but default values are based upon the original LBNL study. In the LBNL study, the occupancy was scaled relative to a daily average maximum occupancy density (people per 1000 ft^2). In VSAT, the user can specify a peak design occupancy density (people per 1000 ft^2) that is used for determining fixed ventilation requirements (no DCV). This same design occupancy density is used as the scaling factor for the hourly occupancy schedules. As a result, the original LBNL occupancy schedules were rescaled using the default peak design occupancy densities.

The heat gains and CO_2 generation per person depend upon the type of building (and associated activity). Design internal gains for lights and equipment also depend upon the building and are scaled according to specified average daily minimum and maximum gain fractions. For all of the buildings, the lights and equipment are at their average maximum values whenever the building is occupied and are at their average minimum values at all other times.

Zone thermostat setpoints can be set for both occupied and unoccupied periods. The default occupied setpoints for cooling and heating are 75°F and 70°F, respectively. The default unoccupied setpoints for cooling (setup) and heating (setback) are 85°F and 60°F, respectively. The lights are assumed to come on one hour before people arrive and stay on one hour after they leave. The occupied and unoccupied setpoints follow this same schedule.

Table 1. Office Building Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.58	
Shading Coefficient	0.75	
Area ratio (window/wall)	0.15	
Exterior Wall Construction		
Layers	1" stone R-5.6 insulation R-0.89 airspace 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") 4" lightweight concrete R-12.6 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete Carpet and pad	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	6600	
Wall height, ft	11	
Internal mass, lb/ft^2	25	
Number of stories	1	
Aspect Ratio	0.67	
Ratio of exterior perimeter to floor perimeter	1.0	
Design equipment gains, W/ft^2	0.5	
Design light gains, W/ft^2	1.7	
Ave. daily min. lights/equip. gain fraction	0.2	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	250	
CO_2 people generation, L/min-person	0.33	
Design occupancy for vent., people/1000 ft^2	7	
Design ventilation, cfm/person	20	
Average weekday peak occucpancy, ft^2/person	470	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-7 8 9 10-16 17 18-24	Values 0.0 0.33 0.66 1.0 0.5 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-8 9 10-12 12-13 13-24	Values 0.0 0.15 0.2 0.15 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-12	Value 1.0

Table 2. Restaurant Dining Area Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.53	
Shading Coefficient	0.8	
Area ratio (window/wall)	0.15	
Exterior Wall Construction		
Layers	3" face brick ½" plywood R-4.9 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.2 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	4" heavyweight concrete Carpet and pad	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	5250	
Wall height, ft	10	
Internal mass, lb/ft^2	25	
Number of stories	1	
Aspect Ratio	1.0	
Ratio of exterior perimeter to floor perimeter	0.75	
Design equipment gains, W/ft^2	0.0	
Design light gains, W/ft^2	2.0	
Ave. daily min. lights/equip. gain fraction	0.2	
Ave. daily max. lights/equip. gain fraction	1.0	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	275	
CO_2 people generation, L/min-person	0.35	
Design occupancy for vent., people/1000 ft^2	30	
Design ventilation, cfm/person	20	
Average weekday peak occucpancy, ft^2/person	50	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-6 7-12 13-24	Values 0.0 0.2,0.3,0.1,0.05,0.2,0.5 0.5,0.4,0.2,0.05,0.1,0.4, 0.6,0.5,0.4,0.2,0.1,0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-6 7-12 13-24	Values 0.0 0.3,0.4,0.5,0.2,0.2,0.3 0.5,0.5,0.5,0.35,0.25, 0.5,0.8,0.8,0.7,0.4,0.2, 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.5 1.0

Table 3. Retail Store Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.5	
Shading Coefficient	0.76	
Area ratio (window/wall)	0.15	
Exterior Wall Construction		
Layers	8" lightweight concrete R-4.8 insulation R-0.89 airspace 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") 1.25" lightweight concrete R-12 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	4" lightweight concrete Carpet and pad	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	80,000	
Wall height, ft	15	
Internal mass, lb/ft^2	25	
Number of stories	2	
Aspect Ratio	0.5	
Ratio of exterior perimeter to floor perimeter	1.0	
Design equipment gains, W/ft^2	0.4	
Design light gains, W/ft^2	1.6	
Ave. daily min. lights/equip. gain fraction	0.2	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	250	
CO_2 people generation, L/min-person	0.33	
Design occupancy for vent., people/1000 ft^2	25	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	390	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-7 8 9 10-20 21 22-24	Values 0.0 0.33 0.66 1.0 0.5 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-7 8 9 10-20 21 22-24	Values 0.0 0.33 0.66 1.0 0.5 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-12	Value 1.0

Table 4. School Class Wing Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8" concrete block R-5.7 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.3 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	9600	
Internal mass, lb/ft^2	25	
Wall height, ft	10	
Number of stories	2	
Aspect Ratio	0.5	
Ratio of exterior perimeter to floor perimeter	0.875	
Design equipment gains, W/ft^2	0.3	
Design light gains, W/ft^2	2.2	
Ave. daily min. lights/equip. gain fraction	0.1	
Ave. daily max. lights/equip. gain fraction	0.95	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	200	
CO_2 people generation, L/min-person	0.3	
Design occupancy for vent., people/1000 ft^2	25	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	50	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-6 7 8-11 12-15 16 17 18 19-21 22-24	Values 0.0 0.1 0.9 0.8 0.45 0.15 0.05 0.33 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-9 10-13 14-24	Value 0.0 0.1 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.5 1.0

Table 5. School Gymnasium Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8" concrete block R-5.7 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.3 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	7500	
Internal mass, lb/ft^2	25	
Wall height, ft	32	
Number of stories	1	
Aspect Ratio	0.86	
Ratio of exterior perimeter to floor perimeter	0.86	
Design equipment gains, W/ft^2	0.2	
Design light gains, W/ft^2	0.65	
Ave. daily min. lights/equip. gain fraction	0.0	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	550	
CO_2 people generation, L/min-person	0.55	
Design occupancy for vent., people/1000 ft^2	30	
Design ventilation, cfm/person	20	
Average weekday peak occucpancy, ft^2/person	180	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-7 8-15 16-24	Value 0.0 1.0 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-24	Value 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.1 1.0

Table 6.   School Library Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8” concrete block R-5.7 insulation 5/8” gypsum	
Roof Construction		
Layers	Built-up roof (3/8”) ¾” plywood R-13.3 insulation R-0.92 airspace ½” acoustic tile	
Floor		
Layers	6” heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	1500	
Internal mass, lb/ft^2	25	
Wall height, ft	10	
Number of stories	1	
Aspect Ratio	0.2	
Ratio of exterior perimeter to floor perimeter	0.75	
Design equipment gains, W/ft^2	0.4	
Design light gains, W/ft^2	1.5	
Ave. daily min. lights/equip. gain fraction	0.1	
Ave. daily max. lights/equip. gain fraction	0.95	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	250	
CO_2 people generation, L/min-person	0.33	
Design occupancy for vent., people/1000 ft^2	20	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	100	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-6 7 8-11 12-15 16 17 18 19-21 22-24	Value 0.0 0.1 0.9 0.8 0.45 0.15 0.05 0.33 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-9 10-13 14-24	Value 0.0 0.1 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.5 1.0

Table 7. School Auditorium Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8" concrete block R-5.7 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.3 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	6000	
Internal mass, lb/ft^2	25	
Wall height, ft	32	
Number of stories	1	
Aspect Ratio	0.64	
Ratio of exterior perimeter to floor perimeter	0.85	
Design equipment gains, W/ft^2	0.2	
Design light gains, W/ft^2	0.8	
Ave. daily min. lights/equip. gain fraction	0.0	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	200	
CO_2 people generation, L/min-person	0.3	
Design occupancy for vent., people/1000 ft^2	150	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	100	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-9 10-11 12 13-14 15-24	Values 0.0 0.75 0.2 0.75 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-24	Value 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.1 1.0

Table 8. Construction Material Properties

	Conductivity (Btu/h*ft*F)	Density (lb/ft^3)	Specific Heat (Btu/lb*F)
stone	1.0416	140	0.20
light concrete	0.2083	80	0.20
heavy concrete	1.0417	140	0.20
built-up roof	0.0939	70	0.35
face brick	0.7576	130	0.22
acoustic tile	0.033	18	0.32
gypsum	0.0926	50	0.20
	Resistance (h*ft^2*F/Btu)		
3/4" plywood	0.93703		
1/2" plywood	0.62469		
carpet and pad	2.08		
inside air	0.67		
outside air	0.33		

2.3 Model Validation

The prototypical buildings were chosen to give representative building loads in order to determine if particular building types will benefit more or less from the ventilation strategies under examination. Absolute model predictions are not the goal but rather the impact of ventilation strategies on savings compared to a baseline. Even so, it is very important that the building load predictions have representative dynamics and absolute load levels. In order to validate predictions of VSAT, results have been compared with predictions of the TYPE 56 building model within TRNSYS (2000). This model has been validated with detailed measurements and through comparison with other accepted building load calculation programs.

The TYPE 56 is a very detailed model that is built up from individual descriptions of wall layers, windows, internal gains, schedules, etc. The user enters all pertinent information into a “front-end” program called *PRE-BID.* This program assimilates all the information into four different files that are used by the TYPE 56 component for generating the specific building loads and ultimately the total building load.

Two building prototypes were chosen as case studies to validate the building loads portion of VSAT. Identical construction properties, schedules, internal gains and weather data for each case study were entered into the TYPE 56 and VSAT models for comparison.

2.3.1 TYPE 56 and VSAT Building Model Assumptions

The TYPE 56 building type predicts the thermal behavior of a building having multiple zones. To determine zone heating and cooling requirements, an “energy rate” method is employed. The user specifies the zone setpoints for heating and cooling with any added setup or setback control schedules. If the floating zone temperature is less than the heating setpoint, then heating is required or if the calculated zone temperature is greater than the cooling setpoint, then cooling is required. Otherwise, the zone temperature is floating and the zone sensible cooling and heating requirement are zero. Unlimited equipment capacity was assumed in the TYPE 56 for purposes of validating the building model in the VSAT.

Walls are modeled in the TYPE 56 using a transfer function method that is equivalent to the approach used in VSAT with a large number of resistors and capacitors. The primary differences between the building model in VSAT and the TYPE 56 are related to the way that solar and long-wave radiation are handled. The solar transmittance for windows is calculated as a function of window properties and solar incidence angle as opposed to the use of a constant shading coefficient employed within VSAT. The solar radiation that is transmitted through windows is distributed to all surfaces in the zone according to the following relation

$$f_j = \frac{\alpha_j \cdot A_j}{\sum_{j=1}^{surfaces} \alpha_j * A_j} \tag{2.30}$$

where f_j is the fraction of transmitted radiation that is absorbed on surface j, A_j is the area of surface j, α_j is the solar absorptance of surface j. In contrast, VSAT distributes all of the transmitted solar radiation to the floor with an even heat flux. It's difficult to say which approach is best, since both are simplifications and the actual solar distribution depends upon the specific geometry of the room and time.

Long-wave radiation exchange between surfaces within the zone is handled in the TYPE 56 using an effective zone surface temperature termed the star temperature. The zone air is coupled to the surface temperatures and star temperature through convective resistances. In contrast, VSAT uses a combined convective and radiative heat transfer coefficient that couples the surface temperatures to the zone air temperature. In both models, surfaces are assumed to be black with respect to long-wave radiation.

Long-wave radiation exchange between outside surfaces and the atmosphere is considered explicitly in the TYPE 56. Radiation occurs between the surface temperatures and an effective temperature that depends upon the surface orientation. The effective temperature is determined as

$$T_{r,o} = (1 - f_{sky}) * T_o + f_{sky} * T_{sky} \tag{2.31}$$

where f_{sky} is the view factor between the surface and the sky, T_o is the outside air temperature, and T_{sky} is a sky temperature that depends upon the air temperature and cloud cover. In contrast, VSAT uses a combined convective and radiative heat transfer coefficient, which is equivalent to assuming that the effective temperature for long-wave radiation is equal to the outside air temperature. In both models, surfaces are assumed to be black with respect to long-wave radiation.

2.3.2 Case Study Description

Two case study descriptions were simulated and compared in VSAT and TRNSYS. The prototypical office and restaurant (see Tables 1 and 2) were both modeled in Madison, WI and San Diego, CA. Only sensible zone loads were considered, not including ventilation.

In VSAT, combined convective and radiation coefficients were utilized for the inside and outside air of 1.5 Btu/hr-ft^2-F and 3.0 Btu/hr-ft^2-F, respectively. Since long-wave radiation is handled explicitly in the TYPE 56, convective heat transfer coefficients need to be specified for the inside and outside air. Convective heat transfer coefficients that result in approximately the combined coefficients used in VSAT were found to be 1.25 Btu/hr-ft^2-F and 2.75 Btu/hr-ft^2-F and were used within the TYPE 56.

The TYPE 56 estimates U-Values for windows based upon the glass properties. For a single pane glass, the U-Value is about 1.0 Btu/hr-ft^2-F. In order to realize the specified overall R-values for the windows used in VSAT, the outside and inside convective heat transfer coefficients were set to 2.3 Btu/hr-ft^2-F and 6.8 Btu/hr-ft^2-F for windows within the TYPE 56.

In order to distribute transmitted solar radiation to the floor only, the solar absorptances of all inside walls were set to zero in the TYPE 56 and the floor solar absorptance was set equal to unity. Finally, the sky temperature used by the TYPE 56 was set equal to the ambient temperature.

2.3.3 Results for Constant Temperature Setpoints

As a first step, cooling and heating loads were evaluated for a constant temperature setpoint of 70°F (21.11°C). This eliminates any transients due to return from night setup and setback. Figure 6 shows hourly heating load comparisons for the office and restaurant over two days in January. VSAT predicts the correct transients and peak load. The relative differences are largest when the loads are smallest at night. Similar results are shown for two days of cooling load predictions in Figure 7.

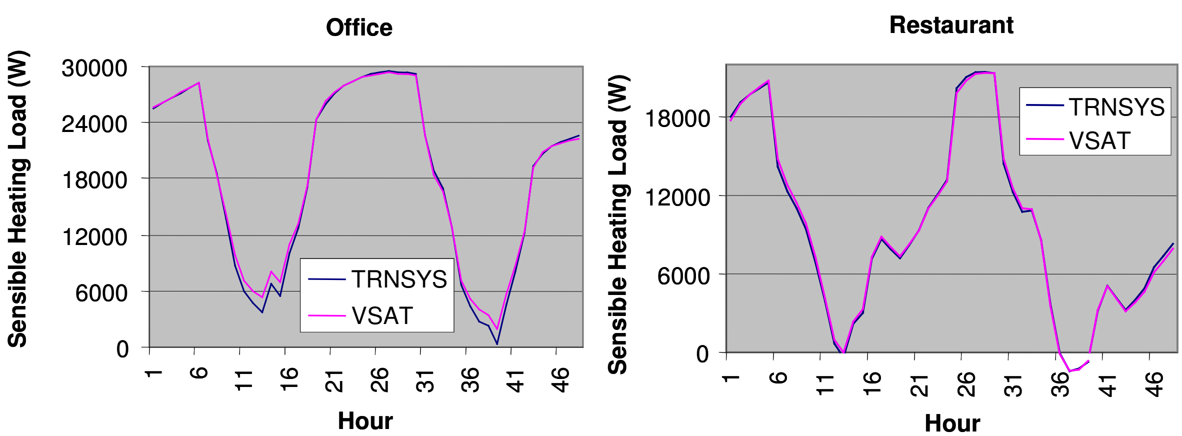


Figure 6. Hourly zone heating loads for constant setpoints (Jan. 9 – 10, Madison, WI)

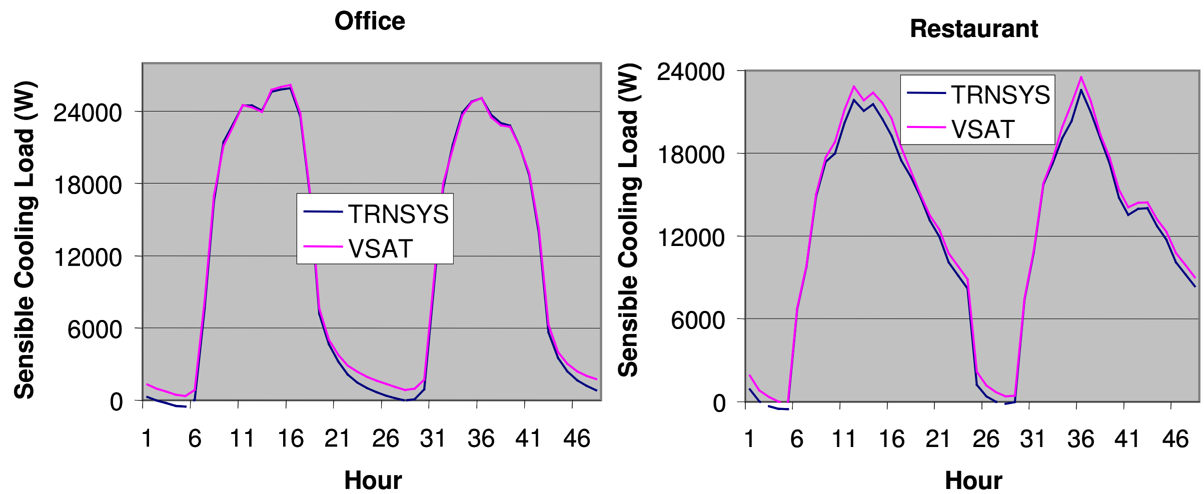


Figure 7. Hourly zone sensible cooling loads for constant setpoints (June 9– 10, San Diego, CA)

Figure 8 and Figure 9 give monthly comparisons for sensible heating and cooling loads. In general, VSAT tends to slightly underpredict heating loads and overpredict cooling loads. This may be due to differences in the manner in which solar radiation transmitted through windows is handle. Overall, the monthly loads are within 5%.

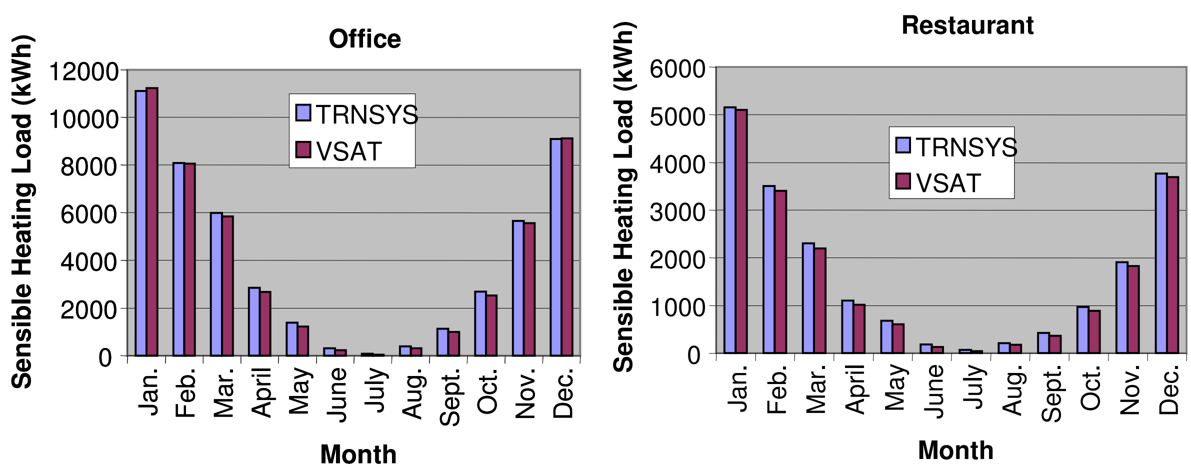


Figure 8. Monthly zone heating loads for constant setpoints (Madison, WI)

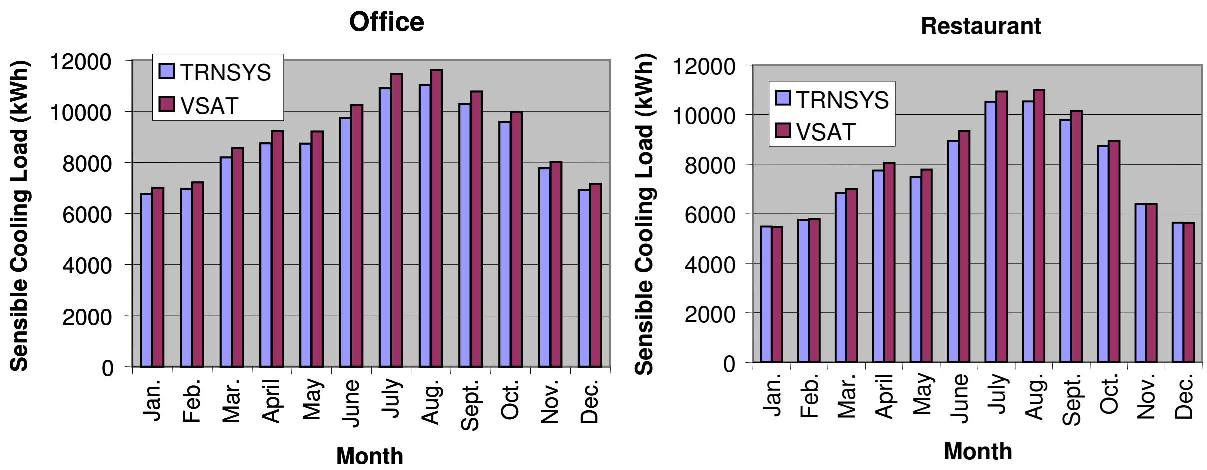


Figure 9. Monthly zone sensible cooling loads for constant setpoints (San Diego, CA)

2.3.4 Results for Night Setback/ Setup Control

The use of a night setback/setup thermostat results in significant dynamics at the start of the occupied period that are not encountered with constant setpoints. Results were generated using both the TYPE 56 and VSAT with night setup for cooling and night setback for heating. For cooling, the occupied period setpoint temperature was 75°F (23.89°C) and the unoccupied setpoint (night setup) temperature was 85°F (29.44°C). For heating, the occupied setpoint was 70°F (21.11°C) and the unoccupied setpoint (night setback) temperature was 60°F (15.56°C). Figure 10 shows sample hourly heat requirements and hourly average zone temperatures for the office in Madison. For both models, there is a large "spike" in the heating requirements when the setpoint returns to the occupied value at 7 am (one hour prior to occupancy). However, the spike is much larger for VSAT than for TRNSYS. This difference is due to differences in the way that zone temperature setpoint adjustsments are handled in the two models. VSAT models a true step change in the setpoint at 7 am, whereas TRNSYS assumes a linear variation in setpoint over the course of the hour from 7 am to 8 am. This difference is apparent in the zone temperature results in Figure 10. Similar results were obtained for the restaurant.

Figure 11 shows similar results for cooling in Madison. Once again, VSAT exaggerates the effect of return from night setup on the cooling loads because it assumes a pure step change in the temperature. Figure 11 also shows that both TRNSYS and VSAT predict similar floating temperatures during the setup (nighttime) period.

Figure 12 shows monthly heating and sensible cooling loads for the office in Madison with night setback/setup control. VSAT tends to overpredict the integrated loads by about 5%. This is partly due to the overprediction of loads at the onset of the return from night setback/setup.

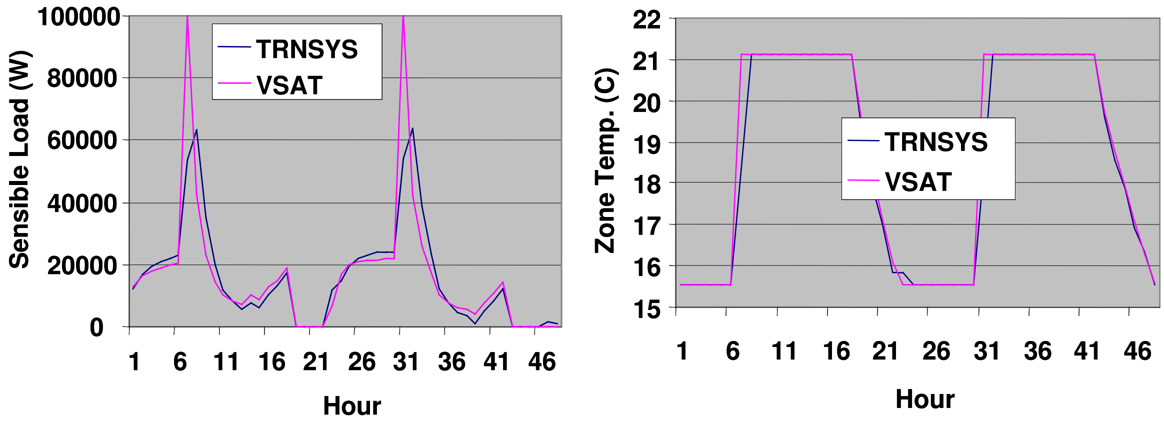


Figure 10. Hourly zone heating loads for the office with night setback (Jan. 9 – 10, Madison, WI)

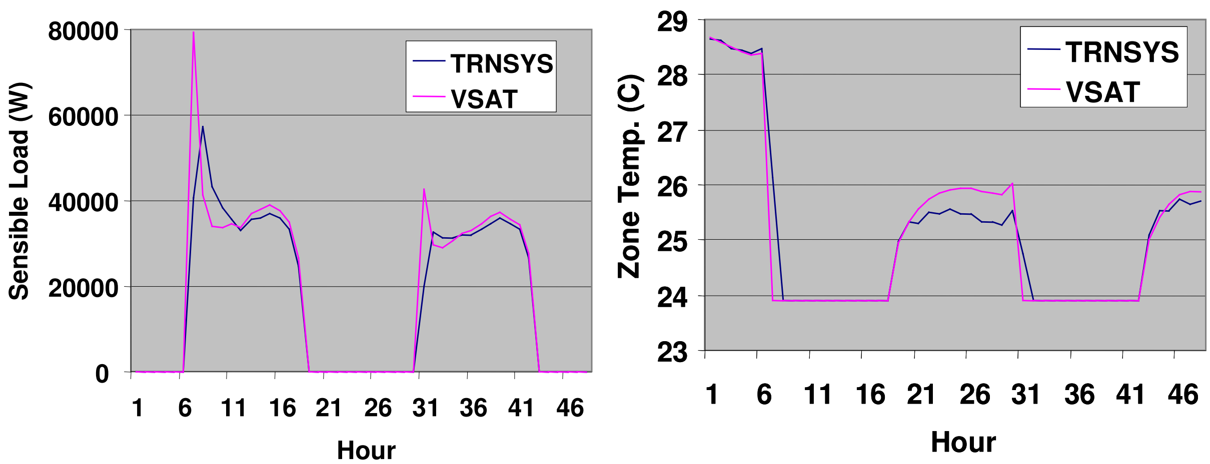


Figure 11. Hourly zone cooling loads for the office with night setup (June 9 – 10, Madison, WI)

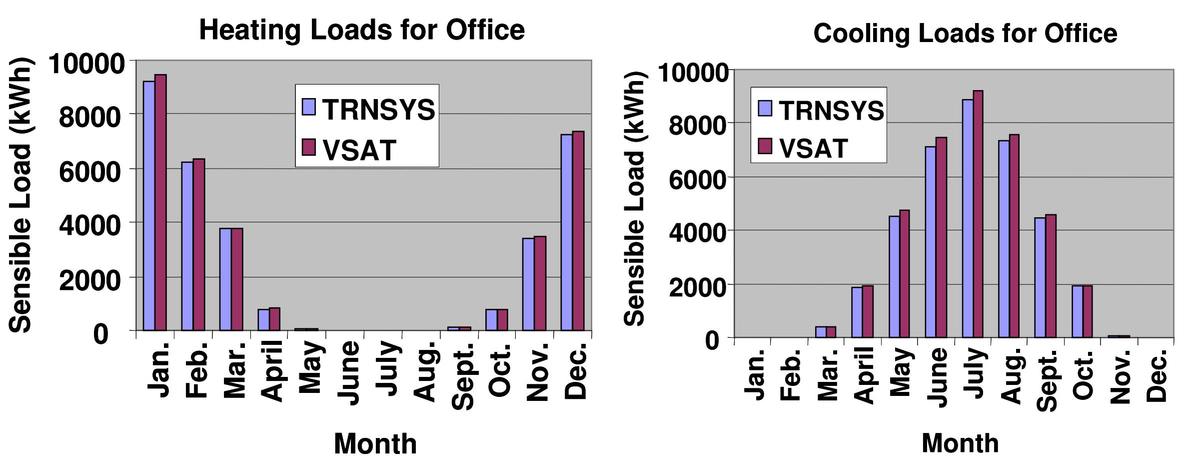


Figure 12. Monthly zone heating and sensible cooling loads for the office with night setup/setup (Madison, WI)

2.3.5 Conclusions

The TYPE 56 building component in TRNSYS is more detailed and accurate in predicting building loads than VSAT. However, for the purposes of comparing different ventilation techniques, this level of detail is not required. Except for return from night setback or setup, VSAT predicts very reasonable transients and overall load levels. Furthermore, VSAT is computationally much more efficient than the TYPE 56, which will facilitate large parametric studies involving many locations and system parameters. The issue of large peak loads at return from night setback or setup will be investigated and VSAT will be modified to predict more reasonable load requirements.

SECTION 3: HEATING AND COOLING EQUIPMENT MODELS

The primary cooling and heating are provided by unitary equipment incorporating a vapor compression air conditioner, a gas or electric heater, and a supply fan. In addition, rotary air-to-air enthalpy exchangers or heat pump heat recovery units can be used to reduce ventilation loads for the primary equipment. Figure 13 depicts a rooftop unit in combination with a heat pump heat recovery unit operating in cooling mode. Ventilation air is cooled and dehumidified by the heat recovery unit prior to mixing with return air from the zone. The mixed air is further cooled and dehumidified (when necessary) by the primary evaporator of the rooftop unit. Heat is rejected to the building exhaust air from the condenser of the recovery unit. The heat pump contains an exhaust fan. In addition, an optional supply fan is used if necessary to provide the proper ventilation air.

In heating mode, refrigerant flow within the heat pump is changed so that the exhaust air stream is cooled (the condenser becomes an evaporator) and heat is rejected to the ventilation air (the evaporator becomes a condenser). The preheated air is then mixed with return air. Although not shown in Figure 13, a gas or electric heater is located after the evaporator to provide additional heating of the supply air when necessary.

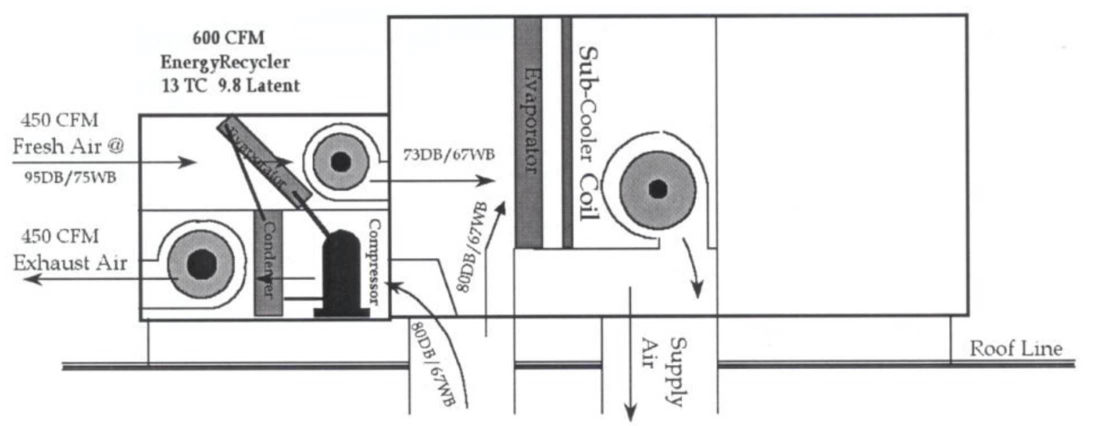


Figure 13. Rooftop air conditioner with heat pump heat recovery unit (cooling mode)

An alternative to heat pump heat recovery is an enthalpy exchanger. Figure 14 depicts a rotary air-to-air enthalpy exchanger considered in VSAT. The device consists of a revolving cylinder filled with an air-permeable medium having a large internal surface area that incorporates a desiccant material. Adjacent supply and exhaust air streams each flow through the exchanger in a counter-flow direction. Sensible heat is transferred as the wheel acquires heat from the hot air stream and releases it to the cold air stream. Moisture is adsorbed from the high humidity air stream to the desiccant material and desorbed into the low humidity air stream. In cooling mode, warm and moist ventilation air is cooled and dehumidified and exhaust air is warmed and humidified. In heating mode, cool and dry air is heated and humidified and exhaust air is cooled and dehumidified.

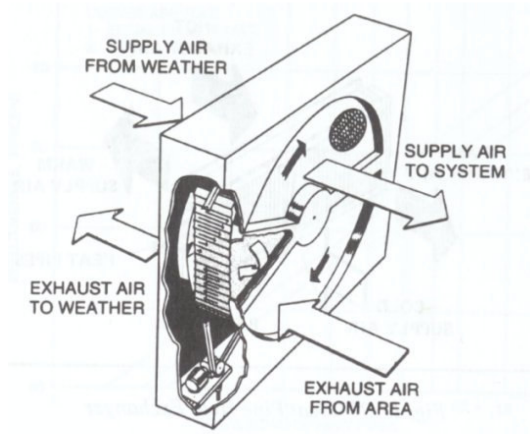


Figure 14. Rotary air-to-air enthalpy exchanger

This section describes the models used for the primary cooling, heating, and heat recovery equipment. Different efficiency equipment can be specified, since this may affect the economics of alternative ventilation strategies. Control strategies for this equipment, along with the description of ventilation control strategies are given in Section 4.

3.1 Vapor Compression System Modeling

Both the primary air conditioning and heat pump heat recovery units utilize a basic vapor compression cycle consisting of a compressor, evaporator coil, expansion valve and condenser coil. Both of these devices are modeled using an approach similar to that incorporated in ASHRAE's *HVAC Toolkit* (Brandemuehl et al., 1993). The model for the primary air conditioner utilizes prototypical performance characteristics, which are scaled according to the capacity requirements and efficiency at design conditions. The characteristics of the heat pump heat recovery unit are based upon measurements obtained from the manufacturer and from tests conducted at the Herrick Labs, which are also scaled for different applications.

3.1.1 Mathematical Description

Steady-State Capacity and COP

The total capacity (cooling or heating), $\dot{Q}_{cap}$, and coefficient of performance, COP, are calculated by applying correction factors to values specified at rating conditions. The correction factors include the effects of air temperature entering the condenser ($T_{c,i}$), evaporator entering wet bulb temperature ($T_{e,wb,i}$) or dry bulb temperature ($T_{e,i}$) and air flow rate ($\dot{m}$). For the case where moisture is removed from the air flowing over the evaporator, the capacity and COP are calculated using the following relations

$$\dot{Q}_{cap} = \dot{Q}_{cap,rat} \cdot f_{cap,t}(T_{e,wb,i}, T_{c,i}) \cdot f_{cap,m}(\dot{m} / \dot{m}_{rat}) \tag{3.1}$$

$$\frac{1}{COP_{cap}} = \frac{1}{COP_{rat}} \cdot f_{COP,t}(T_{e,wb,i}, T_{c,i}) \cdot f_{COP,m}(\dot{m} / \dot{m}_{rat}) \tag{3.2}$$

where $\dot{Q}_{cap}$ and COP_{cap} are the capacity and COP for the unit in steady state with the current operating conditions, $\dot{Q}_{cap,rat}$ and COP_{rat} are the capacity and COP at specified rating conditions, $f_{cap,t}$ is the capacity correction factor based on temperature, $f_{cap,m}$ is the capacity correction factor based on air mass flowrate, $f_{COP,t}$ is the COP correction factor based on temperature, and $f_{COP,m}$ is the COP correction factor based on air mass flowrate. The COP is defined as the ratio of the cooling or heating capacity to the power input. For the primary cooling equipment, the power includes both the compressor and condenser fan, but not the evaporator fan. For the heat pump heat recovery unit, the power includes only the compressor. For either type of equipment, the capacity (cooling or heating) does not include the effect of the supply air fan.

For the primary cooling equipment, the inlet wet bulb temperature to the evaporator is associated with the mixed air condition (mixture of outside and return air) and the inlet condenser temperature is the dry bulb ambient temperature (T_a). The air mass flow rate used within the correlations is the flow rate over the evaporator coil. The air flow rate for the condenser is assumed to be the value at the rating condition.

For the heat pump heat recovery unit, the air flow rate used within the correlations is the ventilation flow rate, which is assumed to be equal for the evaporator and condenser (ventilation and exhaust streams considered to have equal flow rates). For the heat pump recovery unit operating in a cooling mode, the inlet wet bulb to the evaporator is the ambient wet bulb temperature (T_{wb}) and the inlet condenser temperature is the return air temperature from the zone (T_z). During heating mode for the heat pump heat recovery unit, the inlet condenser air temperature is the ambient dry bulb temperature and the inlet condition to the evaporator is the state of air returning from the zone. Since the room air is relatively cool and dry, moisture is not generally condensed as the exhaust air flows over the heat pump evaporator. Therefore, the return air dry bulb temperature (T_z) is used in place of the wet bulb temperature for this case.

The correction factors are based upon correlations of the following form.

$$f_{cap,t}(T_{e,wb,i},T_{c,i}) = a_1 + b_1 \cdot T_{e,wb,i} + c_1 \cdot T_{e,wb,i}^2 + d_1 \cdot T_{c,i} + e_1 \cdot T_{c,i}^2 + f_1 \cdot T_{e,wb,i} \cdot T_{c,i} \tag{3.3}$$

$$f_{COP,t}(T_{e,wb,i},T_{c,i}) = a_2 + b_2 \cdot T_{e,wb,i} + c_2 \cdot T_{e,wb,i}^2 + d_2 \cdot T_{c,i} + e_2 \cdot T_{c,i}^2 + f_2 \cdot T_{e,wb,i} \cdot T_{c,i} \tag{3.4}$$

$$f_{cap,m}(\dot{m}/\dot{m}_{rat}) = a_3 + (\dot{m}/\dot{m}_{rat}) \cdot (b_3 + (\dot{m}/\dot{m}_{rat}) \cdot (c_3 + d_3(\dot{m}/\dot{m}_{rat}))) \tag{3.5}$$

$$f_{COP,m}(\dot{m}/\dot{m}_{rat}) = a_4 + (\dot{m}/\dot{m}_{rat}) \cdot (b_4 + (\dot{m}/\dot{m}_{rat}) \cdot (c_4 + d_4(\dot{m}/\dot{m}_{rat}))) \tag{3.6}$$

Different coefficients are used in equations 3.3 – 3.6 for three different cases: 1) primary cooling unit, 2) heat pump heat recovery operating in a cooling mode, and 3) heat pump heat recovery operating in heating mode. For the primary cooling, the coefficients are from the DOE 2.1E building simulation program. For the heat pump heat recovery unit, the coefficients were determined using performance data as described in a later section.

For cooling, the evaporator inlet air is not always humid enough to result in moisture condensation. In this case, unit performance depends upon inlet evaporator dry bulb rather than wet bulb temperature. However, the correlations developed in terms of wet bulb should provide accurate predictions as long as the correct inlet dry bulb is used and the inlet humidity is set to a value where condensation just begins. This point represents the end of the range where the correlations apply (i.e., the correlation should apply at the point dehumidification begins to occur). Performance is independent of humidity for lower values. Therefore, if the moisture condensation is found not to occur (see section on sensible heat ratio), then the inlet humidity ratio is adjusted until the point where moisture condensation just beings (sensible heat ratio of one). The air inlet wet bulb temperature associated with the actual dry bulb temperature and this fictitious humidity is then used as the evaporator inlet condition for the capacity and COP correlations.

Sensible Heat Ratio

The model for cooling capacity allows determination of the leaving enthalpy using an energy balance, but not the leaving temperature or humidity. A model for moisture removal is utilized that incorporates the concept of a bypass factor (*BF*). The bypass factor approach considers two different air streams flowing across the evaporator. One air stream is in close proximity to the coil surface and exits the evaporator as saturated air at the effective temperature of the coil surface and the other air stream is away from the coil and assumed to remain at the entering air condition. Since the air close to the coil is allowed to come into equilibrium with the effective surface temperature at a saturated condition, then the effective surface temperature must be the dewpoint of inlet air. As a result, it is termed the apparatus dewpoint temperature, T_{adp}.

Mass and energy balances on both air streams give the following

$$\dot{m} = \dot{m}_{app} + \dot{m}_{byp} \tag{3.7}$$

$$\dot{m}\omega_{e,o} = \dot{m}_{app}\omega_{adp} + \dot{m}_{byp}\omega_{e,i} \tag{3.8}$$

$$\dot{m}h_{e,o} = \dot{m}_{app}h_{adp} + \dot{m}_{byp}h_{e,i} \tag{3.9}$$

where $\dot{m}$ is the total air mass flow rate, $\dot{m}_{app}$ is the air mass flow rate near the coil, $\dot{m}_{byp}$ is the air mass flow rate away from the coil (bypass), $h_{e,i}$ and $h_{e,o}$ are the evaporator inlet and outlet air enthalpy, and $\omega_{e,i}$ and $\omega_{e,o}$ are the evaporator inlet and outlet humidity ratio.

The bypass factor is defined as the ratio of the bypass flow to the total flow. With this definition and equations 3.7 – 3.9, the bypass factor can be related to the operating conditions according to

$$BF = \frac{\dot{m}_{byp}}{\dot{m}} = \frac{h_{e,o} - h_{adp}}{h_{e,i} - h_{adp}} = \frac{\omega_{e,o} - \omega_{adp}}{\omega_{e,i} - \omega_{adp}} \tag{3.10}$$

For a given bypass factor (*BF*), equation 3.10 indicates that on a psychrometric chart the outlet air state (h_{eo}, ω_{eo}) is on a straight line that connects the inlet state with the apparatus dewpoint. This is depicted in Figure 15. The larger the bypass factor the closer the outlet state is to the inlet state.

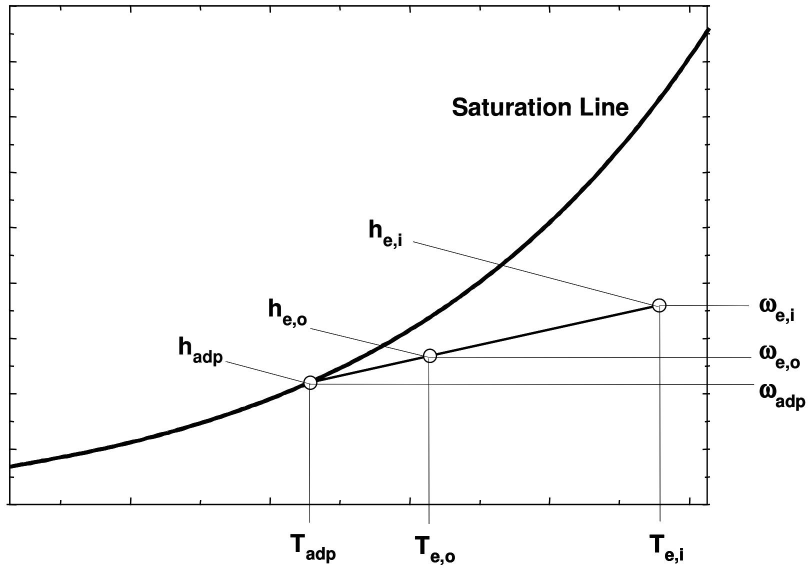


Figure 15. Psychrometric depiction of evaporator air process

The bypass factor can be determined from the heat transfer characteristics of a specific evaporator coil. The bypass factor is estimated from

$$BF = e^{-NTU} \tag{3.11}$$

$$NTU = \frac{UA}{\dot{m} \cdot C_{pm}} \approx \frac{NTU_{rated}}{\left(\dot{m}/\dot{m}_{rated}\right)} \tag{3.12}$$

and where NTU is the number of transfer units, UA is the air-side conductance of the evaporator coil, C_{pm} *is the* specific heat of moist air, and NTU_{rated} is the value of NTU at the rated flow rate. The right-hand form of equation 3.12 employs the assumption that the conductance does not change with air flow rate. NTU_{rated} can be determined from rated performance since the bypass factor can be determined from entering and leaving conditions. Then, the bypass factor is estimated as a function of air flow using equations 3.11 and 3.12.

The outlet air enthalpy for the evaporator operating at steady state is first determined using an energy balance with the known entering enthalpy and the cooling capacity determined as described in the previous section. For a given bypass factor and inlet and outlet enthalpy, the saturated air enthalpy corresponding to the apparatus dewpoint is determined from equation 3.10 as

$$h_{adp} = h_{e,i} - \frac{h_{e,i} - h_{e,o}}{1 - BF} \tag{3.13}$$

The apparatus dewpoint temperature and saturated humidity ratio are determined using psychrometric relationships for a relative humidity of 100% and an air enthalpy of h_{adp}. Then, the outlet air humidity ratio is determined from equation 3.10 as

$$\omega_{e,o} = BF \cdot \omega_{e,i} + (1 - BF) \cdot \omega_{adp} \tag{3.14}$$

Since the outlet state lies on the locus of point connecting the inlet and apparatus dewpoint conditions (see Figure 15), the sensible heat ratio (*SHR*) can be determined as

$$SHR = \frac{h(T_{e,i}, \omega_{adp}) - h_{adp}}{h_{e,i} - h_{adp}} \tag{3.15}$$

where *SHR* is the ratio of the sensible cooling capacity to the total cooling capacity.

If the calculated value of *SHR* is greater then unity, then moisture condensation does not occur and *SHR* is unity. In this case, the inlet humidity ratio is adjusted until the point where *SHR* = 1. The air inlet wet bulb temperature associated with the actual dry bulb temperature and this fictitious humidity is then used as the evaporator inlet condition for the capacity and COP correlations given in the previous section.

Compressor Power Consumption

When there is a cooling requirement for the primary equipment, the compressor(s) and condenser fan(s) cycle on and off to maintain the zone temperature at the cooling setpoint. VSAT utilizes one-hour timesteps and yet the equipment must generally cycle on and off at smaller time intervals. The fraction of the hour that the equipment must operate in order to meet the load is assumed to be equal to the part-load ratio (*PLR*), which is the ratio of the average hourly equipment cooling requirement ($\dot{Q}_c$) to the steady-state capacity ($\dot{Q}_{cap}$) of the equipment or

$$PLR = \frac{\dot{Q}_c}{\dot{Q}_{cap}} \tag{3.16}$$

There are energy losses associated with cycling primarily due to the loss of the pressure differential between the condenser and evaporator when the unit shuts down. The compressor must re-establish the steady-state evaporator and condenser pressures to achieve the steady-state capacity whenever the unit turns on. These pressures equilibrate very quickly after the unit is shut down. The effect of cycling on power consumption is considered through the use of a part-load factor (*PLF*). For any given hour, the average power consumption of the compressor and condenser fan are calculated as

$$\dot{W}_c = PLF \cdot \frac{\dot{Q}_{cap}}{COP_{cap}} \tag{3.17}$$

where *PLF* is ratio of the average power to the full-load power consumption. *PLF* is determined in terms of *PLR* using the following correlation from DOE 2.1E.

$$PLF = a_5 + PLR \cdot (b_5 + PLR \cdot (c_5 + d_5 \cdot PLR)) \tag{3.18}$$

For the heat pump heat recycler, both the ventilation (optional) and exhaust fans operate continuously during the occupied period and do not cycle with the compressor. As a result, the correlations presented for COP_{cap} only include the compressor. For this equipment, the compressor power is determined with equation 3.17.

3.1.2 Prototypical Rooftop Air Conditioner Characteristics

The correlations for the primary rooftop cooling equipment were taken from DOE 2.1E. In VSAT, the rated cooling capacity in equation 3.1 is determined based upon the peak cooling requirements associated with the building, ventilation system, and location (see sizing section). The rated flow rate is 450 cfm/ton. The user can choose between three different rated COPs corresponding to EERs of 8, 10, 12. The default is an EER of 12. The actual evaporator air flow rate when the unit is operating can be set by the user, but the default is 350 cfm/ton.

Figure 16 shows the variation in the temperature-dependent capacity and COP correction factors as a function of condenser air inlet temperature and evaporator air inlet wet bulb temperature for the prototypical rooftop air conditioner. The values were determined with equations 3.3 and 3.4 using the coefficients given in Table 9. The cooling capacity and COP vary by about a factor of two over the range of interest. The maximum capacity and COP (minimum $f_{COP,t}$) occur at a low condenser inlet temperature and high evaporator inlet wet bulb temperature.

Figure 17 shows the mass flow rate-dependent capacity and COP correction factors as a function of the ratio of the supply air flow rate to the rated flow rate. The values were determined with equations 3.5 and 3.6 using the coefficients given in Table 10. Over the range of interest, the impact of supply air flow on COP is relatively small. The COP decreases by only about 5% when the flow is 50% of the design flow. The sensitivity of cooling capacity to changes in flow rate is greater than for COP and the effect becomes more important at low flow rates.

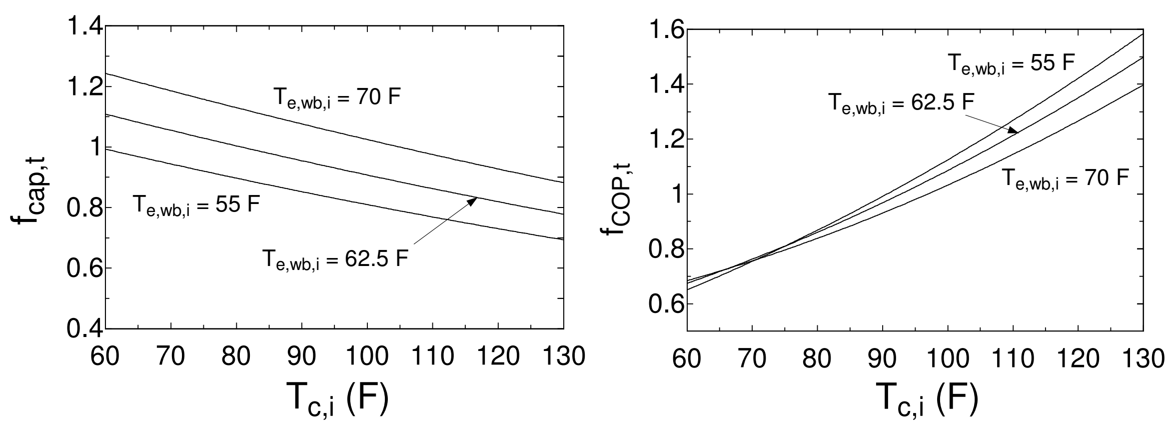


Figure 16. Temperature-dependent capacity and COP correction factors for prototypical rooftop air conditioner

Table 9: Coefficients of temperature-dependent capacity and COP correction factor correlations for the prototypical rooftop air conditioner

Coefficient	Value	Units
a_1	0.8740302	-
b_1	-0.0011416	F^{-1}
c_1	0.0001711	F^{-2}
d_1	-0.0029570	F^{-1}
e_1	0.0000102	F^{-2}
f_1	-0.0000592	F^{-2}
a_2	-1.0639310	-
b_2	0.0306584	F^{-1}
c_2	-0.0001269	F^{-2}
d_2	0.0154213	F^{-1}
e_2	0.0000497	F^{-2}
f_2	-0.0002096	F^{-2}

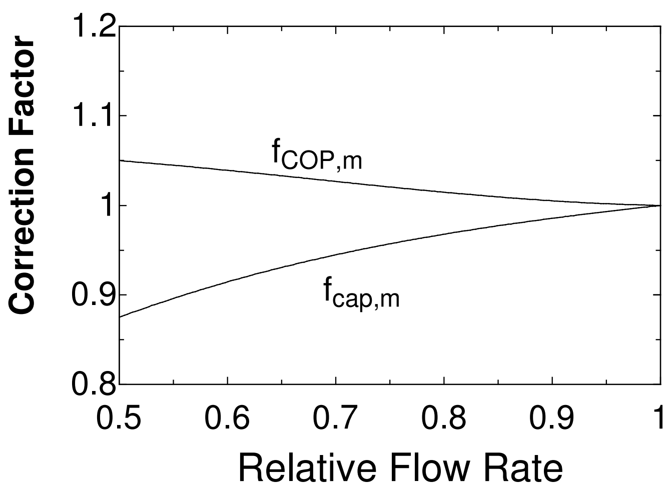


Figure 17. Flow rate-dependent capacity and COP correction factors for prototypical rooftop air conditioner

Table 10: Coefficients of mass flow rate-dependent capacity and COP correction factor correlations for the prototypical rooftop air conditioner

Coefficient	Value
a_3	0.4727859
b_3	1.2433414
c_3	-1.0387055
d_3	0.3225781
a_4	1.0079484
b_4	0.3454413
c_4	-0.6922891
d_4	0.3388994

Figure 18 shows *PLF* as a function of *PLR* determined using the correlation of equation

3.18 with coefficients given in Table 11. Also shown in this plot is a line for constant COP ($PLF = PLR$). The impact of cycling on power consumption is relatively small for part-load ratios greater than about 30%. The deviation from constant COP becomes very significant below a PLR of 0.2.

Table 11: Coefficients of part-load factor correlations

Coefficient	Value
a_5	0.2012301
b_5	-0.0312175
c_5	1.9504979
d_5	-1.1205105

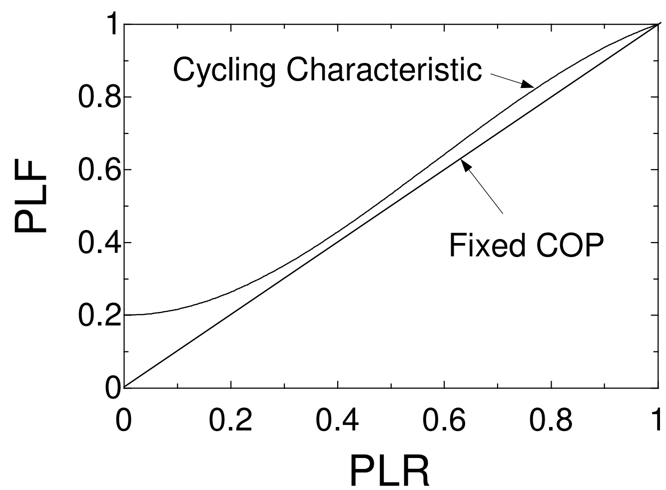


Figure 18. Part-load factor correlation

The user can specify a sensible heat ratio (SHR) at the rating condition. This value is used along with the rated flow rate per unit cooling capacity (cfm/ton) and the rated operating conditions to determine the rated bypass factor (BF) and NTU. The bypass factor is then corrected for the actual flow rate using equations 3.11 and 3.12. Standard ARI rating conditions are assumed: condenser inlet temperature of 95 F, evaporator inlet temperature of 80 F, and evaporator inlet wet bulb temperature of 67 F. The default value for the rated SHR is 0.75. For the prototypical unit with these specifications, the rated bypass factor is 0.261 and the rated NTU is 1.35.

3.1.3 Heat Pump Heat Recovery Unit (Energy Recycler®)

The heat pump heat recovery unit is modeled using a very similar approach as for the primary air conditioner except that equation 3.2 is replaced with

$$COP_{cap} = COP_{rat} \cdot f_{COP,t}(T_{e,wb,i}, T_{c,i}) \cdot f_{COP,m}(\dot{m} / \dot{m}_{rat}) \tag{3.19}$$

This form resulted in better correlation of data. Coefficients of equations 3.3 - 3.6 were determined using manufacturer's data and tests conducted at the Herrick Labs. The laboratory tests provided data beyond the range available from the manufacturer. The rating conditions for the heat pump were taken from suggested rating points given in the manufacturer's data for both cooling and heating modes (Carrier, 1999).

Cooling Mode

For the unit considered, the rated air supply flow rate for cooling mode is 533 cfm/ton (1000 cfm rated supply air divided by 22.5 MBtu/hr gross cooling capacity). Rated air conditions are 75°F condenser air inlet dry bulb temperature, 95°F evaporator air inlet dry bulb temperature and 75°F evaporator air inlet wet bulb temperature. For the unit tested at this rating point, the total capacity is 22.5 MBtu/hr, COP is 4.515 and SHR is 0.902. The Energy Recycler is not available at different EERs, thus only one performance characteristic is available for analysis. The coil heat transfer units (NTUs) parameter at the rated condition is 1.08 and the rated bypass factor is 0.34.

Figure 19 shows the variation of temperature dependent correction factors for total cooling capacity and COP as a function of condenser air inlet temperature and evaporator air inlet wet bulb temperature. The correction factors were determined from equations 3.3 and 3.4 using the coefficients in Table 12.

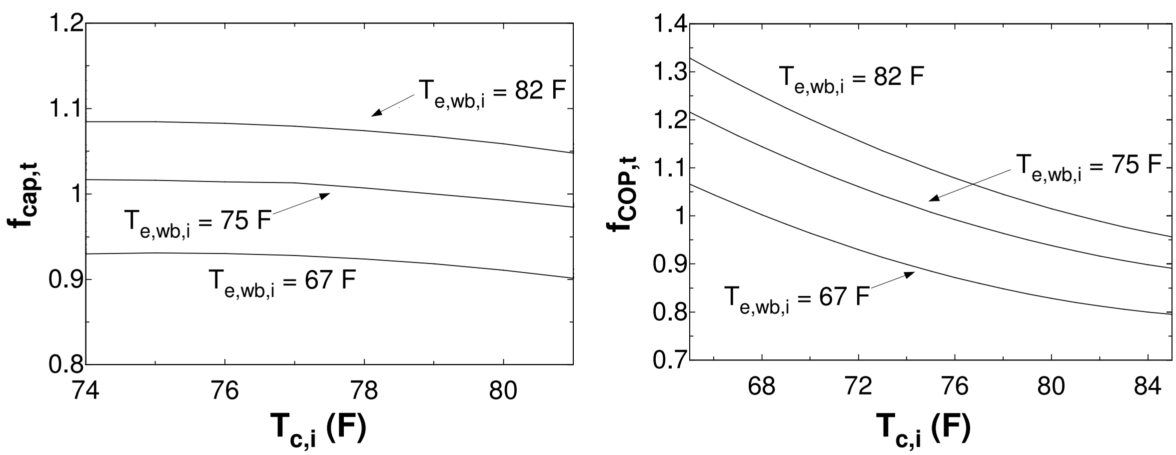


Figure 19. Temperature-dependent capacity and COP correction factors for heat pump heat recovery unit – cooling mode

Table 12: Coefficients of temperature-dependent capacity and COP correction factor correlations for the heat pump heat recovery unit – cooling mode

Coefficient	Value	Units
a_1	-6.758	-
b_1	0.0946	F^{-1}
c_1	-0.000223	F^{-2}
d_1	0.09721	F^{-1}
e_1	-0.0003967	F^{-2}
f_1	-0.0005549	F^{-2}
a_2	0.8402	-
b_2	0.06599	F^{-1}
c_2	-0.0001786	F^{-2}
d_2	-0.0592	F^{-1}
e_2	0.0004547	F^{-2}
f_2	-0.0003368	F^{-2}

Figure 20 shows the variation of mass dependent correction factors for total cooling capacity and COP as a function of the flow rate relative to the rated flow rate. The correction factors were determined from equations 3.5 and 3.6 using the coefficients in Table 13. The impact of flow rate on performance is much more significant than for the primary air conditioner because both condenser and evaporator flow rate change (not just evaporator flow rate).

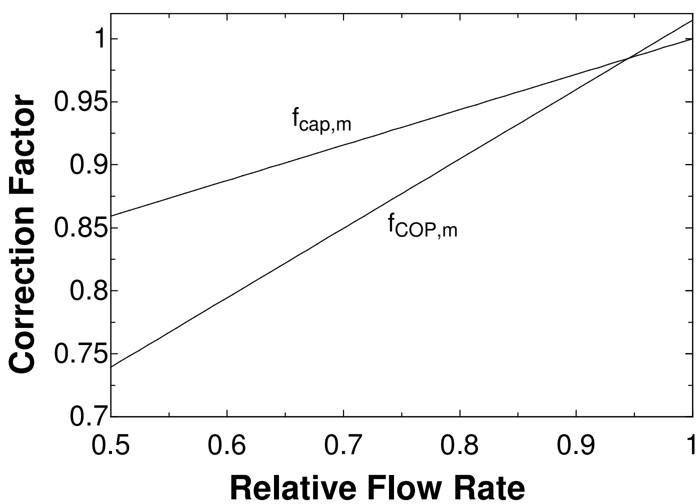


Figure 20. Flow rate-dependent capacity and COP correction factors for heat pump heat recovery unit – cooling mode

Table 13: Coefficients of mass flow rate-dependent capacity and COP correction factor correlations for heat pump heat recovery unit – cooling mode

Coefficient	Value
a_3	0.7187
b_3	0.2813
c_3	0.0
d_3	0.0
a_4	0.4639
b_4	0.5509
c_4	0.0
d_4	0.0

Heating Mode

For the unit considered, the rated air supply flow rate for heating mode is 540 cfm/ton (1000 cfm rated supply air divided by 22.2 MBtu/hr gross heating capacity). Rated air conditions are 70°F evaporator air inlet temperature and 33°F condenser air inlet temperature. The total capacity is 22.2 MBtu/hr and COP is 7.425 at this rating point.

Figure 21 shows the variation of the temperature dependent correction factors for total heating capacity and COP as a function of evaporator (return) air inlet temperature and condenser air inlet temperature. The correction factors were determined from equations 3.3 and 3.4 using the coefficients in Table 14. Total heating capacity and COP increase as the evaporator inlet temperature increases and condenser inlet temperature decreases. The maximum capacity for heating is thus experienced at higher air evaporating temperatures and lower condenser inlet temperatures.

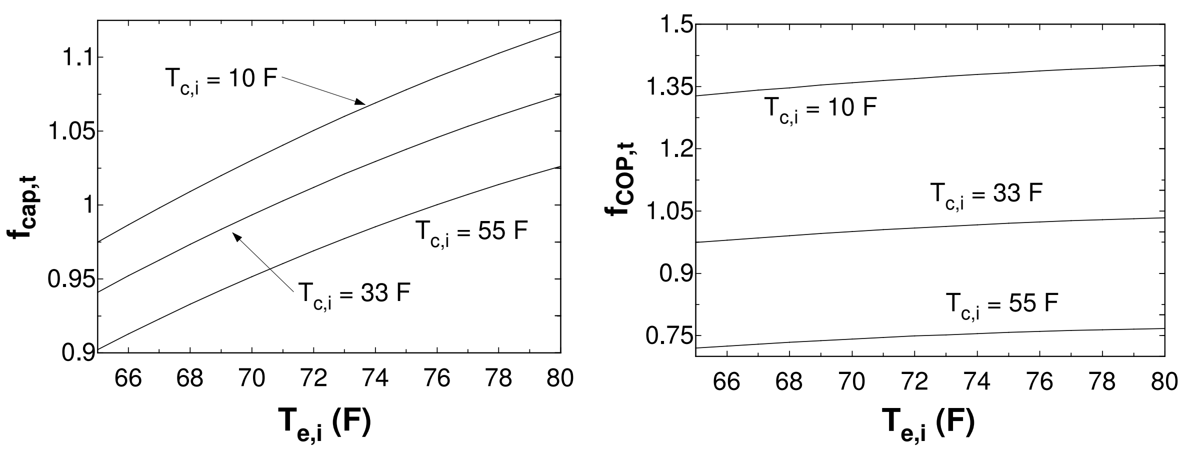


Figure 21. Temperature-dependent capacity and COP correction factors for heat pump heat recovery unit – heating mode

Table 14: Coefficients of temperature-dependent capacity and COP correction factor correlations for the heat pump heat recovery unit – heating mode

Coefficient	Value	Units
a_1	-0.4831	-
b_1	0.0006157	F^{-1}
c_1	-0.000006376	F^{-2}
d_1	0.03305	F^{-1}
e_1	-0.0001604	F^{-2}
f_1	-0.0000279	F^{-2}
a_2	0.4873	-
b_2	-0.01648	F^{-1}
c_2	0.00008504	F^{-2}
d_2	0.02423	F^{-1}
e_2	-0.0001307	F^{-2}
f_2	-0.00003938	F^{-2}

Figure 22 shows the variation of mass dependent correction factors for total heating capacity and COP as a function of the relative flow rate. The correction factors were determined from equations 3.5 and 3.6 using the Energy Recycler coefficients in Table 15.

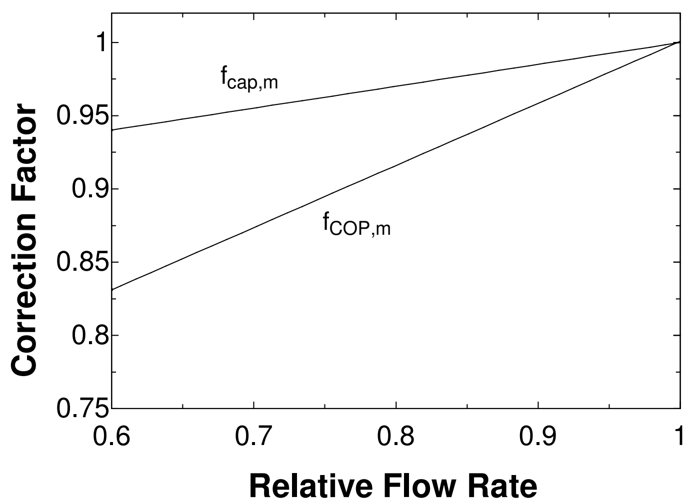


Figure 22. Flow rate-dependent capacity and COP correction factors for heat pump heat recovery unit – heating mode

Table 15: Coefficients of mass flow rate-dependent capacity and COP correction factor correlations for heat pump heat recovery unit – heating mode

Coefficient	Value
a_3	0.8505
b_3	.1495
c_3	0.0
d_3	0.0
a_4	0.5768
b_4	0.424
c_4	0.0
d_4	0.0

3.2 Primary Heater

The primary heater incorporated within the rooftop unit can be either gas or electric. For electric heat, the power consumption at any time is assumed to be equal to the heating requirement for any hour. For a gas heater, gas costs are based upon the primary fuel energy consumption integrated over the billing period (therms). For any time, the rate of primary fuel energy consumption is calculated as

$$\dot{Q}_F = \frac{\dot{Q}_h}{\eta_F} \tag{3.20}$$

where $\dot{Q}_h$ is the heating requirement for the heating and η_f is the heater efficiency. The efficiency is assumed to be constant. The user can choose between three different efficiencies of 0.65, 0.80, and 0.95. The default efficiency is 0.95

3.3 Enthalpy Exchanger

3.3.1 Mathematical Description

The model for enthalpy exchangers that is incorporated within VSAT was developed by Stiesch et al. (1995) and Klein et al. (1990). Both of these studies incorporate the use of temperature, humidity, and enthalpy effectiveness defined as

$$\varepsilon_T = \frac{T_v - T_a}{T_z - T_a} \tag{3.21}$$

$$\varepsilon_\omega = \frac{\omega_v - \omega_a}{\omega_z - \omega_a} \tag{3.22}$$

$$\varepsilon_h = \frac{h_v - h_a}{h_z - h_a} \tag{3.23}$$

where ε is effectiveness, T is temperature, ω is humidity ratio, h is enthalpy, and the subscripts a, v, and z refer to conditions associated with the ambient air, ventilation air leaving the enthalpy exchanger, and return air from the zone, respectively.

For known values of effectiveness, equations 3.21 – 3.23 are used to estimate ventilation stream conditions in terms of ambient and zone air conditions. As the effectiveness values go to one, the ventilation temperature, humidity, and enthalpy approach the conditions of the return air. In general, the effectiveness increases as the speed of the wheel increases for given air flow rates.

Klein et al. (1990) used detailed numerical studies and found that for balanced flow rates, a Lewis number of one, and at high rotation speeds, the temperature, humidity, and enthalpy effectiveness for enthalpy exchangers are equal and can be estimated in terms of the number of transfer units as

$$\varepsilon_T = \varepsilon_\omega = \varepsilon_h = \frac{NTU}{NTU + 2} \tag{3.24}$$

where NTU is defined as

$$NTU = \frac{hA_s}{\dot{m}_{vent} C_{pm}} \tag{3.25}$$

and h is the heat transfer coefficient and A_s is the total surface area of the exchanger.

Stiesch et al. (1995) correlated temperature and enthalpy effectiveness as a function of rotation speeds, where the results were generated from detailed simulations. The correlations are of the form

$$\varepsilon_T = \frac{NTU}{NTU + 2} \cdot (1 - \exp[a_T \cdot \Gamma^2 + b_T \cdot \Gamma]) \tag{3.26}$$

$$\varepsilon_h = \frac{NTU}{NTU + 2} \cdot (1 - \exp[a_h \cdot \Gamma^3 + b_h \cdot \Gamma^2 + c_h \cdot \Gamma]) \tag{3.27}$$

where the *a, b,* and *c* coefficients are empirical factors that depend upon ambient temperature and Γ is a dimensionless rotation speed defined as

$$\Gamma = \frac{M_m / \dot{m}_{vent}}{t_r} \tag{3.28}$$

and where t_r is the time required for one exchanger rotation, M_m is the mass of the dry matrix, and $\dot{m}_{vent}$ is the ventilation flow rate.

Equations 3.26 and 3.27 tend to approach the limiting case result of equation 3.24 for dimensionless rotation speeds greater than about 3. Well-designed enthalpy exchangers would tend to operate at higher speeds. However, it may be necessary to operate at lower speeds to maintain a fixed ventilation supply air temperature under feedback control conditions.

Feedback control of the wheel speed is initiated under two situations: 1) the ambient air temperature is below 55 F and the ventilation stream outlet air temperature rises above 55 F or 2) the exhaust stream outlet air temperature falls below a freeze setpoint. The control logic incorporated in VSAT is based upon typical practice (Semco, 2002).

If the ambient temperature is below 55 F and the ventilation stream outlet air temperature falls would rise above 55 F (at full speed), then the wheel speed is modulated below the maximum speed to maintain an outlet temperature of 55 F. This limits preheating of the ventilation stream under conditions where cooling may be required. The temperature effectiveness necessary to achieve this condition is calculated as

$$\varepsilon_{T,vent,sp} = \frac{T_{vent,sp} - T_a}{T_z - T_a} \tag{3.29}$$

where $T_{vent,sp}$ is the setpoint temperature (55 F) for the ventilation supply air. Under low ambient conditions, the ventilation temperature is below 55 F and the wheel operates at full speed.

At low ambient temperatures, water vapor removed from the exhaust stream may condense and freeze. Reducing the speed reduces the effectiveness of the enthalpy exchanger and increases the matrix temperature within the exhaust speed. Freeze protection is initiated in VSAT when the exhaust temperature falls below a specified freeze protection limit. In this case, the exhaust temperature is set equal to the freeze protection limit and the temperature effectiveness necessary to achieve this condition is calculated as

$$\varepsilon_{T,freeze} = \frac{T_z - T_{freeze}}{T_z - T_a} \tag{3.30}$$

where T_{freeze} is the freeze protection limit for the exhaust temperature.

For either feedback control case, the dimensionless rotation speed necessary to achieve the required effectiveness given by equation 3.29 or 3.30 is determined from equation 3.26 using the required temperature effectiveness. Then, the ventilation stream enthalpy is evaluated using equations 3.27 and 3.23.

A frost set point is specified based on winter ambient and zone design conditions as discussed by Semco (2002) and Stiesch (1995). Figure 23 depicts the process on a psychrometric chart. Point A1 represents a low ambient temperature condition, whereas points Z1 and Z2 represent zone conditions with high and low humidities, respectively. For an enthalpy exchanger operating at full speed, the ventilation and exhaust air streams follow processes that are approximately on these lines. For process line Z2-A1, the exhaust air process line never crosses the saturation line and therefore moisture would not condense. However, for process line Z1-A1, moisture condenses at point Z1a for a wheel operating at full speed. In this case, the frost setpoint should be set a temperature greater than the temperature at Z1a.

The frost set point is determined by first estimating the point where the enthalpy exchanger process line (e.g., one Z1-A1) crosses the saturation line (e.g., point Z1a) assuming 1) an ambient condition of 90% relative humidity at the lowest temperature occurring during the year and 2) a zone condition of 35% relative humidity at the heating setpoint. The crossing point is determined numerically and then a 2 C safety factor is added to the result

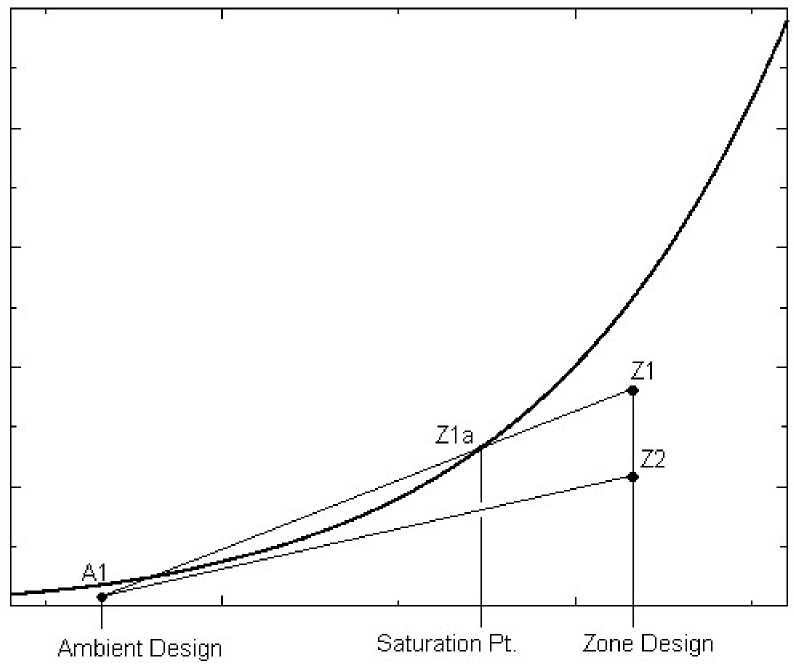


Figure 23. Frost set point determination

.

3.3.2 Prototypical Exchanger Descriptions

The specific correlations developed by Stiesch et al. (1995) were obtained using data from a commercial enthalpy exchanger (Carnes 1989) having a medium constructed of aluminum foil of thickness 0.025 mm coated with a thin, uniform layer of polymer desiccant. The medium has a counter-flow design and is constructed by coiling smooth and corrugated aluminum sheets to produce small triangular air passages. The equivalent hydraulic diameter of the triangular air passages is approximately 1.7 mm and the medium has a length in the direction of flow of 0.2 m. The diameter of the wheel is 1.23 m and the idealized rotational speed is approximately 15 rpm.

The manufacturer gives effectiveness as a function of air face velocity for their designs. At a face velocity of 650 fpm, the effectiveness for heat and mass transfer is about 0.75. From equation 3.24, this results in an NTU of about 6. These values are assumed for the prototypical enthalpy exchanger.

The effectiveness is constant unless the feedback control is initiated as described in the previous subsection. If this occurs, then the empirical factors determined by Stiesch et al. (1995) are used in equations 3.26 and 3.27. These are:

$$a_T = a_{T_1} + \frac{a_{T_2}}{NTU^{a_{T_3}}} \tag{3.31}$$

$$b_T = b_{T_1} + \frac{b_{T_2}}{NTU^{b_{T_3}}} \tag{3.32}$$

$$a_h = a_{h_1} + a_{h_2} \cdot NTU + a_{h_3} \cdot NTU^2 \tag{3.33}$$

$$b_h = b_{h_1} + b_{h_2} \cdot NTU + b_{h_3} \cdot NTU^2 \tag{3.34}$$

$$c_h = c_{h_1} + \frac{c_{h_2}}{NTU^{0.8}} \tag{3.35}$$

where

$$a_{T_1} = 0.002259 - 1.376\times10^{-3} \cdot T_a - 6.91\times10^{-6} \cdot T_a^2$$
$$a_{T_2} = 0.09084 - 3.263\times10^{-4} \cdot T_a + 7.4\times10^{-6} \cdot T_a^2$$
$$a_{T_3} = 0.7388 - 0.01994 \cdot T_a - 3.829\times10^{-4} \cdot T_a^2$$
$$b_{T_1} = -1.007 + 0.0093 \cdot T_a + 2.778\times10^{-4} \cdot T_a^2$$
$$b_{T_2} = -1.533 + 0.02287 \cdot T_a - 2.356\times10^{-4} \cdot T_a^2$$
$$b_{T_3} = 1.111 - 2.667\times10^{-3} \cdot T_a - 1.378\times10^{-4} \cdot T_a^2$$
$$a_{h_1} = 3.381\times10^{-3} - 9.679\times10^{-4} \cdot T_a \qquad \text{for } T_a \le 0°C$$
$$a_{h_1} = 3.381\times10^{-3} - 4.127\times10^{-5} \cdot T_a \qquad \text{for } T_a > 0°C$$
$$a_{h_2} = 5.088\times10^{-4} + 4.89\times10^{-6} \cdot T_a$$
$$a_{h_3} = 5.298\times10^{-6} - 7.652\times10^{-7} \cdot T_a$$
$$b_{h_1} = 6.237\times10^{-3} + 8.827\times10^{-3} \cdot T_a - 6.042\times10^{-4} \cdot T_a^2$$
$$b_{h_2} = -0.02133 + 1.323\times10^{-4} \cdot T_a$$
$$b_{h_3} = 4.908\times10^{-4} + 6.46\times10^{-6} \cdot T_a$$
$$c_{h_1} = -0.4087 + 0.00253 \cdot T_a + 3.34\times10^{-4} \cdot T_a^2$$
$$c_{h_2} = -1.449 + 0.02337 \cdot T_a - 5.578\times10^{-4} \cdot T_a^2$$

SECTION 4: AIR DISTRIBUTION SYSTEM AND CONTROLS

The air distribution system includes ducts, fans, dampers, and controls. A supply fan integrated with the primary cooling/heating equipment provides the flow rate to the zone. A return fan is not considered. The ventilation heat pump heat recovery unit utilizes an exhaust fan and an optional ventilation fan. The ventilation fan is only necessary if the required ventilation flow rate cannot be provided using the primary supply fan. During the occupied period, the fan(s) operate(s) continuously and provide a constant flow rate of air to the zone, while the equipment cycles on and off as necessary to maintain the zone temperature setpoint. During the unoccupied period, the fan(s) cycle(s) on and off with the equipment, but the airflow rate is constant when the system is on.

There are separate heating and cooling setpoints for the zone. If the zone temperature falls between these setpoints, then the temperature is "floating" and no heating or cooling is required. If the zone temperature falls below the heating setpoint, then the heating required to maintain the zone at this temperature is calculated as the zone heating load. The total equipment heating load includes an additional load associated with ventilation. If the zone temperature rises above the cooling setpoint, then the cooling required to maintain the zone at this temperature is calculated as the sensible zone cooling load. The total equipment cooling load includes additional loads associated with ventilation and latent gains within the zone.

When installed, the ventilation heat pump heat recovery unit is only enabled during occupied hours. During unoccupied hours, the primary air conditioner and heater must meet the cooling and heating requirements. In addition, the heat pump will only operate in cooling mode when the ambient temperature is above 68 F.

The enthalpy exchanger operates when the primary fan is on and the ambient temperature is less than 55 F or greater than the return air temperature. When the ambient temperature is between 55 F and the return air temperature, it is assumed that a cooling requirement exists and it is better to bring in cooler ambient air. When the ambient temperature is below 55 F, then a feedback controller adjusts the speed to maintain a ventilation supply air temperature of 55 F. When the ambient temperature is above the return air temperature, then wheel operates at maximum speed.

There are four ventilation control strategies considered in VSAT: fixed ventilation, demand-controlled ventilation, economizer, and night ventilation precooling. When a heat recovery heat exchanger or heat pump is employed within the ventilation stream, then fixed ventilation is assumed. Demand-controlled ventilation is considered both with and without an economizer. Night ventilation is considered with and without an economizer and with and without demand-controlled ventilation.

This section describes modeling of the air distribution components and controls and calculation of the equipment heating and cooling loads.

4.1 Ventilation Flow

4.1.1 Fixed Ventilation

In the absence of demand-controlled ventilation and during occupied mode, the minimum ventilation flow rate is a fixed value and is determined using ASHRAE Standard 62-1999 based upon the design occupancy. Table 1 - Table 7 include ventilation requirements and design occupancies for the prototypical buildings considered in VSAT. Note that in many cases, the average occupancy levels are much lower than the design occupancies used to determine minimum ventilation flow requirements. During unoccupied mode, the minimum ventilation flow is zero and the damper is closed.

4.1.2 Demand-Controlled Ventilation

When demand-controlled ventilation is enabled, a minimum flow rate of ventilation air is determined that will keep the CO_2 concentration in the zone at or below a specified level. The minimum flow rate is calculated assuming a quasi-steady state mass balance on the air within the zone and the ducts, fully-mixed zone air, and a constant ventilation effectiveness that accounts for short-circuiting of ventilation air within the supply to the return duct. With these assumptions, the minimum ventilation flow rate is

$$\dot{m}_{vent,\min} = \min\left(\frac{\dot{C}_{CO_2,gen}}{\eta_v \cdot \left(C_{CO_2,set} - C_{CO_2,amb} \right)}, \dot{m}_{\sup} \right) \tag{4.1}$$

where $\dot{C}_{CO_2,gen}$ is the rate of CO_2 generation within the zone, $C_{CO_2,set}$ is the setpoint for CO_2 concentration in the zone, $C_{CO_2,amb}$ is the ambient CO_2 concentration, and η_v is the ventilation efficiency. The ventilation efficiency is a measure of how effectively the ventilation air removes pollutants from the zone. The default value is 0.85. The user can set values for the zone setpoint and ambient CO_2 concentrations. The default values are 1000 ppm and 350 ppm, respectively.

The CO_2 generation rate is the product of the generation rate per person and the number of occupants at any given time. Table 1 - Table 7 include generation rates per person and default occupancy information for the prototypical buildings considered in VSAT.

4.1.3 Economizer

At any given time, the ventilation flow can be greater than the minimum due to economizer operation. VSAT considers a differential enthalpy economizer. The differential enthalpy economizer is engaged whenever the enthalpy of the ambient air is less than the enthalpy of the air in the return duct and the zone requires cooling.

In economizer mode, the ventilation flow rate is modulated between the minimum and maximum (wide open) values to maintain a specified setpoint for the mixed air temperature supplied to the primary equipment. The default mixed air setpoint is 55 F. During the occupied mode, the economizer will cycle on and off as necessary to maintain the zone temperature setpoint. However, during unoccupied mode, both the economizer and the fan cycle on and off together to maintain the zone temperature. In either case, the average hourly ventilation flow rate when the economizer is enabled is determined as

$$\dot{m}_{vent} = \min\left(\max\left(\dot{m}_{vent,\min}, \dot{m}_{vent,mix}\right), \dot{m}_{vent,z}, \dot{m}_{\sup}\right) \tag{4.2}$$

where $\dot{m}_{vent,mix}$ is the ventilation flow rate necessary to give a mixed air temperature equal to its setpoint and $\dot{m}_{vent,z}$ is the ventilation flow rate that keeps the zone temperature at its setpoint. This logic simulates a perfect economizer controller that requires a call for 1st stage cooling to enable the economizer (and fan during unoccupied mode) and uses damper modulation to maintain a mixed air temperature setpoint.

With the economizer enabled, the ventilation flow rate necessary to maintain the zone temperature at its setpoint is

$$\dot{m}_{vent,z} = \frac{\dot{Q}_{z,c} + \dot{W}_{fan,s}}{C_{pm}\left(T_{z,c} - T_a\right)} \tag{4.3}$$

where $\dot{Q}_{z,c}$ is the zone sensible cooling load, $T_{z,c}$ is the zone temperature setpoint for cooling, and $\dot{W}_{fan,s}$ is the power associated with the primary supply fan.

4.1.4 Night Ventilation Precooling

Whenever the ambient temperature drops below the zone temperature, the ambient air can be used to precool the zone and reduce cooling loads during the next day. However, the next day savings associated with operating the ventilation system at night should be sufficient to offset the cost of operating the fan. In addition, the ambient humidity should be low enough to avoid increased latent loads during the next day and the ambient temperature should be high enough so as to avoid additional heating requirements after occupancy. With these issues in mind, the rules in Table 16 are employed to enable precooling.

Table 16. Rules for Enabling Ventilation Precooling

Rule	Description
$(T_z - T_a) > \Delta T_{on}$	The ambient temperature (T_a) must be less than the zone temperature (T_z) by a threshold (ΔT_{on}) chosen to balance fan operating costs with next day savings.
$T_a > 50$˚F	The ambient temperature must be greater than 50 ˚F to avoid conditions where heating might be required the next day.
$T_{a,dp} < 55$˚F	The ambient dew point ($T_{a,dp}$) must be less than 55 ˚F to avoid conditions where the latent load might increase the next day.
$\Delta t_{occ} < 6$ hours	The time to occupancy (Δt_{occ}) must be less than 6 hours to achieve good storage efficiency.
$N_{heat} > 24$ hours	The number of hours (N_{heat}) since the last call for heating should be greater than 24 hours to lock out precooling in the heating season

When night ventilation precooling is enabled, mechanical cooling is disabled and the ventilation system operates with 100% outside air to precool the zone with a setpoint of 67 ºF. Once the zone temperature reaches 67 ºF, the fan cycles to maintain this setpoint. Just prior to the occupied period, the setpoint for ventilation precooling is raised to 69 ºF. Once the occupied period begins, there are separate setpoints associated cooling provided by the economizer (1st stage cooling) and the packaged air conditioner (2nd stage cooling). The 1st and 2nd stage setpoints are 69 ºF and 75 ºF, respectively. Once the occupied period ends, the zone temperature setpoint is raised to 80 ºF.

The threshold for the zone/ambient temperature difference is determined based upon trading off nighttime fan energy and daytime compressor energy saved. When ventilation precooling is enabled, mechanical cooling is disabled and the zone temperature setpoint is set at 67 F. The damper is fully open and the ventilation flow rate is equal to the primary supply air flow rate. The fan cycles, as necessary, to maintain the zone setpoint. At this point, the temperature difference required to achieve savings is estimated from equation 2.7 as

$$\Delta T_{on} = \frac{\dot{W}_{fan}}{\rho_a c_{pa} \dot{V}_{fan}} \cdot \left(\frac{COP_{nv,occ}}{\eta_s} \cdot \frac{\overline{R}_{unocc}}{\overline{R}_{occ}} + 1 \right) \tag{4.4}$$

Figure 24 shows the breakeven temperature difference as a function of the ratio of unoccupied to occupied energy rates and the ratio of fan power to volumetric flow rate for a storage efficiency of 0.8 and an occupied period COP of 3. For typical values, the threshold varies between about 1 F and 10 F. The breakeven point increases with fan power (i.e., pressure drop) for a given flow rate since the cost of providing a given quantity of precooling increases. The fan power typically varies between about 0.4 and 0.7 W/cfm. The threshold also increases as the ratio between occupied and unoccupied energy rates decreases. Lower occupied period energy costs reduce the savings associated with precooling leading to a larger threshold. For similar reasons, the threshold increases with increasing occupied period COP. For packaged air conditioning equipment, the COP varies between about 2 and 4. Finally, the threshold increases with decreasing storage efficiency as less of the precooling results in cooling load reductions during the occupied period. Storage efficiencies vary between about 0.5 and 0.9.

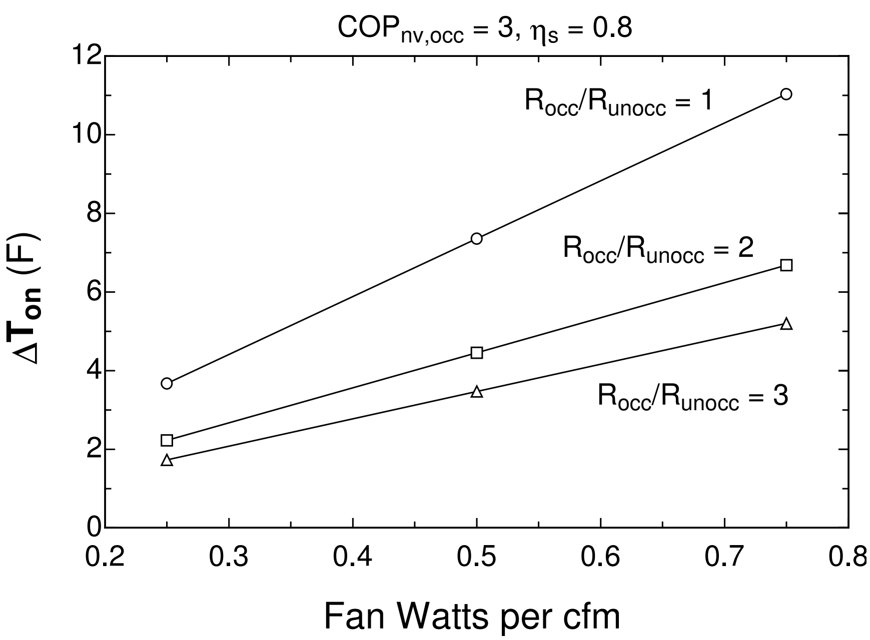


Figure 24. Night Ventilation Breakeven Threshold

4.2 Mixed Air Conditions

The mixed air conditions entering the primary air conditioner and heater are determined from mass and energy balances for adiabatic mixing according to

$$\omega_{mix} = \frac{\dot{m}_{vent}}{\dot{m}_{\sup}}\omega_v + \left(1 - \frac{\dot{m}_{vent}}{\dot{m}_{\sup}}\right)\omega_z \quad (4.4)$$

$$h_{mix} = \frac{\dot{m}_{vent}}{\dot{m}_{\sup}}h_v + \left(1 - \frac{\dot{m}_{vent}}{\dot{m}_{\sup}}\right)h_z \quad (4.5)$$

where ω is humidity ratio, h is air enthalpy, and the subscripts z and v refer to zone return and ventilation air conditions, respectively. For a system with a heat recovery heat exchanger or heat pump, ω_v and h_v are the conditions exiting the device within the ventilation stream. Otherwise, these properties are evaluated at ambient humidity conditions. The mixed air temperature is evaluated with psychrometric property routines in terms of the mixed air humidity and enthalpy or

$$T_{mix} = T(h_{mix}, \omega_{mix}) \quad (4.6)$$

4.3 Equipment Heating Requirements

If the zone requires heating to maintain the temperature at the heating setpoint, then the furnace and/or heat pump heat recovery unit must operate to meet the zone requirements and any additional load associated with ventilation. The furnace and heat pump provide only sensible heating (no humidification). If a heat pump heat recovery unit is employed, then it has the first priority for heating (i.e., 1st stage heating) during occupied mode. During unoccupied mode, the heat pump unit does not operate.

4.3.1 Heat Pump Heat Recovery Unit

From an energy balance on the air within the zone and distribution system, the heating load for the heat pump during occupied mode is

$$\dot{Q}_{hphr,h} = \min\left(\dot{Q}_{z,h} + \dot{m}_{vent}C_{pm}(T_{z,h} - T_a) - \dot{W}_{fan,s} - \dot{W}_{fan,v}, \dot{Q}_{hphr,h,cap}\right) \quad (4.7)$$

where $\dot{Q}_{z,h}$ is the zone heating load, $\dot{m}_{vent}$ is the ventilation flow rate, $T_{z,h}$ is the zone heating temperature setpoint, $\dot{W}_{fan,s}$ is the power associated with the primary supply fan, $\dot{W}_{fan,v}$ is the power associated with the optional ventilation fan for the heat pump, and $\dot{Q}_{hphr,h,cap}$ is the heating capacity associated with the heat pump.

4.3.2 Primary Heater

The heating requirement for the primary heater is

$$\dot{Q}_h = \dot{Q}_{z,h} + \dot{m}_{vent}C_{pm}(T_{z,h} - T_v) - \dot{W}_{fan,s} \quad (4.8)$$

where T_v is the temperature of the ventilation air that is mixed with return air. For a system with a heat recovery heat exchanger or heat pump, T_v is the temperature exiting the device within the ventilation stream. Otherwise, T_v is equal to the ambient temperature.

4.4 Equipment Cooling Requirements

The first priority for cooling (1st stage cooling) is the economizer if it is installed and enabled. If the economizer can meet the sensible zone cooling requirement, then the primary air conditioner does not operate. If a heat pump heat recovery unit is installed, then an economizer is not employed and the heat pump is the first priority for cooling during occupied mode. During unoccupied mode, the heat pump unit does not operate.

4.4.1 Heat Pump Heat Recovery Unit

The sensible cooling requirement for the heat pump is

$$\dot{Q}_{hphr,s,c} = \min\left(\dot{Q}_{s,T}, SHR \cdot \dot{Q}_{hphr,c,cap}\right) \tag{4.9}$$

where $\dot{Q}_{hphr,c,cap}$ is the cooling capacity of the heat pump, SHR is the heat pump sensible heat ratio, and $\dot{Q}_{s,T}$ is the total sensible load determined as

$$\dot{Q}_{s,T} = \dot{Q}_{z,c} + \dot{m}_{vent} C_{pm} (T_a - T_{z,c}) + \dot{W}_{fan,s} + \dot{W}_{fan,v} \tag{4.10}$$

where $\dot{Q}_{z,c}$ is the zone sensible cooling load. The cooling capacity and SHR are evaluated using the ambient and zone return air conditions as inlet conditions for the evaporator and condenser.

The total cooling requirement for the heat pump is

$$\dot{Q}_{hphr,c} = \frac{\dot{Q}_{hphr,s,c}}{SHR} \tag{4.11}$$

4.4.2 Primary Air Conditioner

The sensible cooling requirement for the primary air conditioner is

$$\dot{Q}_{ac,s,c} = \min\left(\dot{Q}_{s,T}, SHR \cdot \dot{Q}_{ac,c,cap}\right) \tag{4.12}$$

where $\dot{Q}_{ac,c,cap}$ is the cooling capacity of the air conditioner, SHR is the air conditioner sensible heat ratio, and $\dot{Q}_{s,T}$ is the total sensible load determined as

$$\dot{Q}_{s,T} = \dot{Q}_{z,c} + \dot{m}_{vent} C_{pm} (T_v - T_{z,c}) + \dot{W}_{fan,s} \tag{4.13}$$

where T_v is the temperature of the ventilation air that is mixed with return air. For a system with a heat recovery heat exchanger or heat pump, T_v is the temperature exiting the device within the ventilation stream. Otherwise, T_v is equal to the ambient temperature.

The cooling capacity and *SHR* are evaluated using the mixed air conditions as described in Section 3. When an economizer is not enabled, the mixed air condition depends on both ventilation and zone return air conditions according to equations 4.4 and 4.5. However, the return air humidity depends on the exit humidity from the air conditioner, which in turn depends on the mixed air condition. A quasi-steady state mass balance for humidity within the air distribution system is used along with an iterative solution to determine the zone and mixed air states and equipment performance. The zone return air humidity ratio must satisfy equations 4.4, 4.5, 4.12, 4.13 and the following equations.

$$(1 - SHR) \cdot \dot{Q}_{ac,c} = \dot{Q}_{p,L} + \dot{m}_{\inf}(\omega_a - \omega_z)h_{fg} + \dot{m}_{vent}(\omega_v - \omega_z)h_{fg} \tag{4.14}$$

$$\dot{Q}_{ac,c} = \frac{\dot{Q}_{ac,s,c}}{SHR} \tag{4.15}$$

where $\dot{Q}_{ac,c}$ is the total equipment cooling requirement, $\dot{Q}_{p,L}$ is the latent load associated with people in the zone, $\dot{m}_{\inf}$ is the infiltration flow rate, ω_a is the ambient humidity ratio, and h_{fg} is the heat of vaporization of water.

4.5 Supply, Ventilation, and Exhaust Fans

Only single-speed air distribution fans are considered in VSAT. For systems without a heat pump heat recovery unit or enthalpy exchanger, only a single supply fan is used for each primary air conditioner. The heat pump heat recovery unit incorporates a fan for the exhaust stream and has an optional fan for the ventilation stream. Enthalpy exchangers typically employ both ventilation and exhaust stream fans to ensure effective purging. For each fan, the fan power is scaled with the volumetric flow according to

$$\dot{W}_{fan,on} = w_f \cdot \dot{V}_{on} \tag{4.16}$$

where $\dot{W}_{fan,on}$ is fan power at steady state, w_f is fan power per unit of volume flow and $\dot{V}_{on}$ is the volumetric flow rate when the fan is operating. The user can specify values for w_f. For the primary supply fans, the default value for w_f is 0.5 W/cfm. For the ventilation and exhaust fans, the default value for w_f is 0.25 W/cfm.

During occupied mode, any of the air distribution fans operate continuously. However, during unoccupied mode, the fans cycle with the heater or primary air conditioner and/or economizer. In this case, the average hourly fan power is calculated as

$$\dot{W}_{fan} = PLR \cdot \dot{W}_{fan,on} \tag{4.17}$$

where *PLR* is the ratio of the average hourly heating or cooling requirement to the heat or cooling capacity. When heating or mechanical cooling is required, then the *PLR* is determined as outlined in Section 3. When cooling is required and the economizer can meet the cooling requirements, then *PLR* is determined as

$$PLR = \frac{\dot{Q}_{z,c}}{\dot{m}_{\sup} C_{pm} (T_{z,c} - T_{mix,econ})} \tag{4.18}$$

where $T_{mix,econ}$ is the mixed air setpoint temperature for the economizer.

4.6 Zone Controls – Call for Heating or Cooling

The first step in evaluating whether heating or cooling is required is to determine the zone temperature if the equipment were off. During unoccupied mode, the supply air fan is off when there is no heating and cooling requirement. In this case, the floating zone temperature is determined by setting $\dot{Q}_z$ to zero in equation 2.27 and solving for the zone temperature. During occupied mode, the fan(s) operate(s) continuously so that ventilation loads and fan energy influence the floating zone temperature. In this case, the zone temperature is determined that satisfies the following equation.

$$\dot{Q}_z + \dot{W}_{fan,s} + \dot{W}_{fan,v} + \dot{m}_{vent} C_{pm} \left(T_a - T_z\right) = 0 \tag{4.19}$$

SECTION 5: WEATHER DATA, SIZING, AND COSTS

5.1 Weather Data

VSAT contains typical meteorological year (TMY2) weather data for 239 US locations and California Climate Zone data for 16 representative zones within California. The data include hourly values of ambient temperature, horizontal radiation, and direct normal radiation. In addition, the user can specify the ambient CO_2 level. The default value is 350 ppm.

The California Climate Zones are shown in Figure 25 and the representative cities for each climate zone (CZ) are given in Table 17. The climate zones are based on energy use, temperature, weather and other factors. They are basically a geographic area that has similar climatic characteristics. The California Energy Commission originally developed weather data for each climate zone by using unmodified (but error-screened) data for a representative city and weather year (representative months from various years). The Energy Commission analyzed weather data from weather stations selected for (1) reliability of data, (2) currency of data, (3) proximity to population centers, and (4) non-duplication of stations within a climate zone.

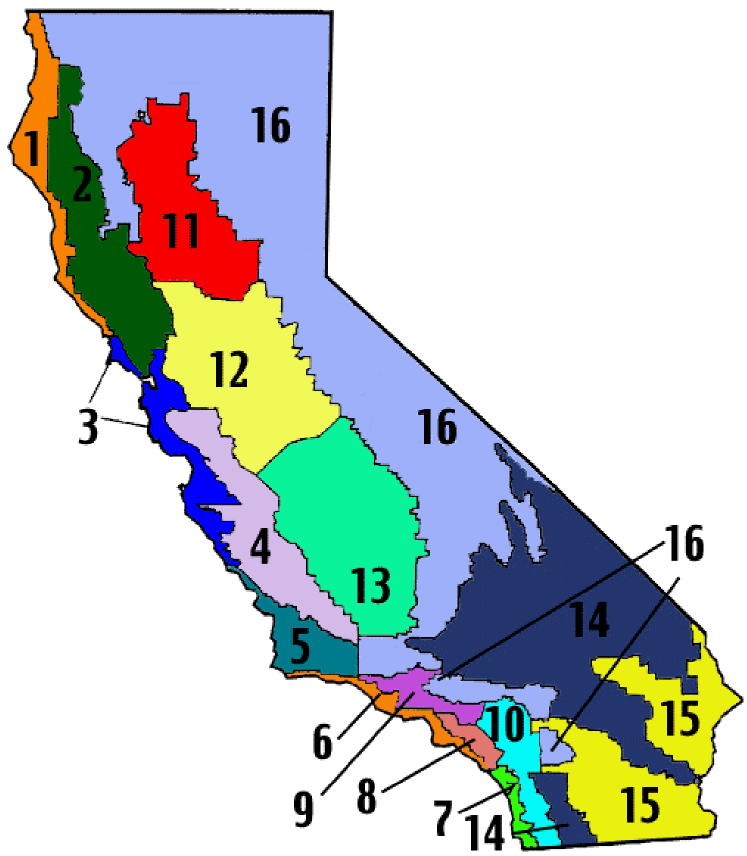


Figure 25. California Climate Zones

Table 17. Cities associated with California Climate Zones

CZ 1: Arcata	CZ 5: Santa Maria	CZ 9: Pasadena	CZ13: Fresno
CZ 2: Santa Rosa	CZ 6: Los Angeles	CZ10: Riverside	CZ14: China Lake
CZ 3: Oakland	CZ 7: San Diego	CZ11: Red Bluff	CZ15: El Centro
CZ 4: Sunnyvale	CZ 8: El Toro	CZ12: Sacramento	CZ16: Mount Shasta

There are two sets of Climate Zone data included in VSAT, the original and a massaged set. In the massage data, the dry bulb temp has been modified in an effort to give the file a better "average" across the entire zone. Because only dry bulb was adjusted, the humidity conditions are affected and therefore the massaged files are not preferred.

5.2 Equipment Sizing

The heating and cooling equipment are automatically sized for a given building and location. The primary heating and cooling equipment are sized assuming no ventilation heat recovery (enthalpy exchanger or heat pump), no economizer, fixed ventilation, and constant zone temperature setpoints (no night setup or setback). The peak sensible heating and cooling requirements are first determined by calculating the hourly zone and ventilation loads throughout the heating and cooling seasons. The heating capacity is set at 1.4 times the peak sensible heating load.

For cooling, the required equipment cooling capacity depends upon the latent load, which depends on ambient and zone humidities and the zone internal latent gains. The required capacity is determined iteratively using the ambient conditions and zone latent gains associated with the peak sensible cooling requirement along with the equipment and air distribution models. The cooling equipment is then oversized by 10%.

The supply air flow rate is determined based upon a specified flow per unit cooling capacity with a default of 350 cfm/ton. The supply fan power is based upon a specified fan power per unit flow rate with a default of 0.5 W/cfm.

The number of rooftop units employed for a given application will influence the economics of the different ventilation strategies. Individual rooftop units require separate enthalpy exchangers, heat pump heat recovery units, economizers, or controllers (demand-controlled ventilation or night ventilation precooling). It will be assumed that rooftop units are available in sizes of 3.5, 5, 7.5, 10, 15, and 20 ton cooling capacities. For a given application and location, the number of individual rooftop units will be based upon the using fewest possible number of units necessary to realize a cooling capacity that is greater than, but within 10% of the target equipment cooling capacity.

The diameter of individual enthalpy exchangers will be scaled so as to achieve a flow velocity of 650 fpm. At this velocity, the exchanger has a constant effectiveness for heat and mass transfer of 0.75 when operated at normal speed.

The heat pump heat recovery unit cooling capacity will be scaled to achieve a flow per unit cooling capacity of 533 cfm/ton based upon the rated cooling capacity and the ventilation flow requirements.

5.3 Costs

VSAT is set up to calculate the simple payback period associated with different ventilation strategies. The alternatives are compared with a base case that has fixed ventilation with no economizer or other ventilation strategy. For any alternative k, the simple payback period is calculated as

$$N_{pb} = \frac{C_k}{S_k} \tag{5.1}$$

where S_k is the annual savings in utility costs associated with the ventilation strategy as compared with the base case and C_k is the installed cost associated with implementing the ventilation strategy.

The annual utility costs associated with operating the HVAC system are calculated according to

$$C_{HVAC} = \sum_{m=1}^{m=12} \left\{ \begin{array}{c} r_{d,on,m} \cdot \dot{W}_{peak,on,m} + r_{d,mid,m} \cdot \dot{W}_{peak,mid,m} + r_{d,off,m} \cdot \dot{W}_{peak,off,m} \\ + \sum_{i=1}^{N_m} \left(r_{e,i,m} \cdot W_{i,m} + r_{g,i,m} \cdot G_{i,m} \right) \end{array} \right\} \tag{5.2}$$

where m is the month, i is the hour, N_m is the number of hours in month m, and for each month m: $r_{d,on,m}$, $r_{d,mid,m}$ and $r_{d,off,m}$ are the utility rates for electricity demand during the on-peak, mid-peak and off-peak periods (\$/kW) and $\dot{W}_{peak,on,m}$, $\dot{W}_{peak,mid,m}$ and $\dot{W}_{peak,off,m}$ are the peak power consumption for the HVAC equipment during the on-peak, mid-peak and off-peak periods; and for each hour i of month m: r_e is the utility rate for electricity usage (\$/kWh), W is the electricity usage (kWh), r_g is the utility rate for natural gas usage (\$/therm), G is the gas usage (therm).

The electricity costs include both energy (\$/kWh) and demand charges (\$/kW) for on-peak, off-peak, and mid-peak periods. Gas energy usage does not vary with time of the day. However, the user can enter different electric and gas rates for summer and winter periods.

The default rates and periods incorporated in VSAT are given in Table 18, Table 19, and Table 20. The default electric utility rates incorporated in VSAT are based upon Pacific, Gas and Electric Company (PG&E) Schedule E-19. The default natural gas rates are based on PG&E Schedule G-NR1.

Table 18. Default time periods for utility rates

PG&E			
Summer:	May 1 - Oct. 31	Winter:	Nov. 1 - April 30
On-Peak	12:00 - 6:00, M - F	On-Peak	N/A
Mid-Peak	8:30 AM - 12:00 & 6:00 PM - 9:30 PM, M - F	Mid-Peak	8:30 AM - 9:30 PM, M - F
Off-Peak	9:30 PM - 8:30 AM, all week	Off-Peak	9:30 PM - 8:30 AM, all week

Table 19. Default natural gas rates in VSAT

PG&E Schedule G-NR1, CA Climate Zones 1, 2, 3, 4, 11, 12, 13, 1	
Summer Season	\$0.67355
Winter Season	\$0.74220

Table 20: Default electric rates in VSAT

PG&E Schedule E-19, CA Climate Zones 1, 2, 3, 4, 5, 11, 12, 13			
Energy Charge - $/kWh			
Summer Season	On-Peak	$0.08773	
	Mid-Peak	$0.05810	
	Off-Peak	$0.05059	
Winter Season	On-Peak	N/A	
	Mid-Peak	$0.06392	
	Off-Peak	$0.05038	
Time Related Demand Charge - $kW			
Summer Season	On-Peak	$13.35	
	Mid-Peak	$3.70	
	Off-Peak	$2.55	
Winter Season	On-Peak	N/A	
	Mid-Peak	$3.65	
	Off-Peak	$2.55	

SECTION 6: SAMPLE RESULTS AND COMPARISONS WITH ENERGY-10

6.1 Sample Results

Figure 26 shows sample hourly results for the Base Case (night setup with no economizer) and with Night Ventilation Precooling for the school class wing within early summer in Climate Zone 10 obtained using the default VSAT utility rates (PG&E E-19 and GNR-1). Night ventilation precooling is enabled during the unoccupied mode when the ambient temperature is sufficiently cooler than the zone temperature. For this example, this occurs during the hour from 11-12:00 pm and continues until the occupied mode begins at 5 am. Prior to occupancy the zone temperature is cooled to around 20˚C. At occupancy, the economizer keeps the zone temperature at a lower economizer setpoint until 8 am when the temperature begins to rise. The temperature reaches the setpoint for mechancial cooling at 11:00 am.

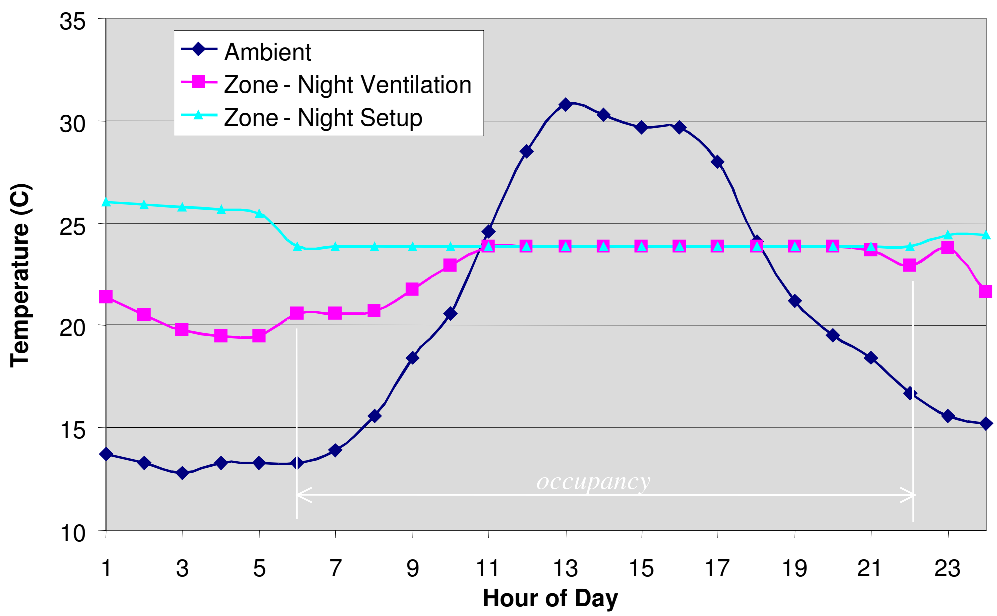


Figure 26. Sample hourly results night ventilation precooling and the base case (class school wing during early summer in Rialto, California).

Figure 27 shows hourly fan and compressor power comparisons for the situation considered for Figure 26. Additional fan energy is utilized during the early morning hours with night ventilation precooling, but this leads to a reduction in compressor energy over much of the day. Part of the savings is due to the low zone setpoint for the economizer, which acts to maintain a cool building thermal mass during the morning hours. For the night ventilation control, mechanical cooling is not needed until 11 am. Clearly, the night ventilation control requires significantly less compressor energy and has slightly lower peak electrical demand at the expense of additional fan energy.

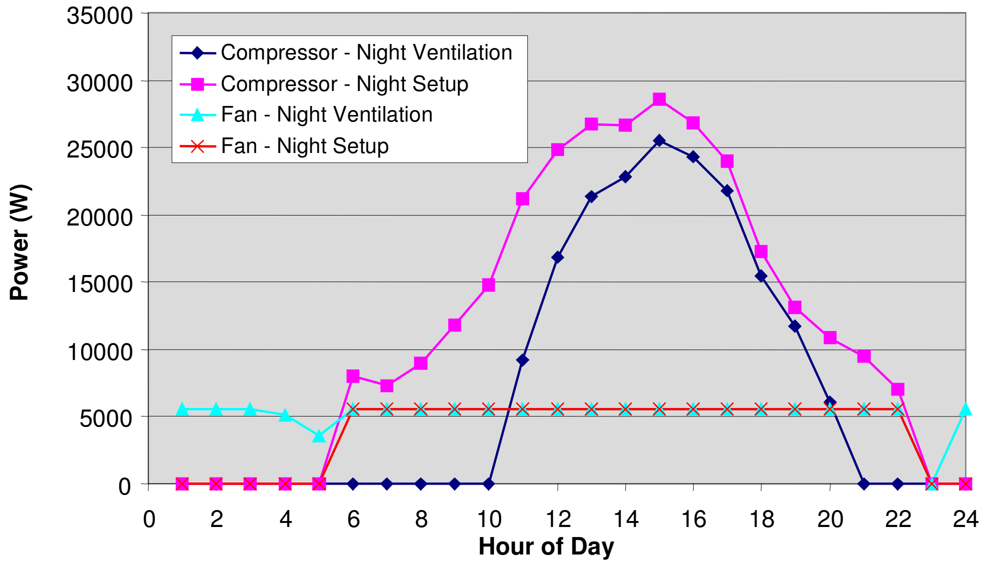


Figure 27: Simulated hourly power for night ventilation precooling and the base case (class school wing during early summer in California Climate Zone 10).

Figure 28 gives annual electrical energy usage for the class school wing in California Climate Zone 10 for three ventilation strategies: 1) a base case with a night setup thermostat, 2) case 1 with the addition of a differential enthalpy-based economizer, and 3) case 2 with the addition of the night ventilation precooling algorithm. Compared to the base case, the economizer results in a savings in compressor energy of 17.4%. The combined compressor and fan savings are about 11.1%. Compared to the economizer, the addition of the night ventilation algorithm leads to an additional savings of about 14.0% in compressor energy. However, the fan energy increases by about 14.2%, and because the compressor energy is the major consumption, the energy saved in compressor is more than the additional comsumption by fan, the combined savings about 2.6% is achieved compared to the economizer only.

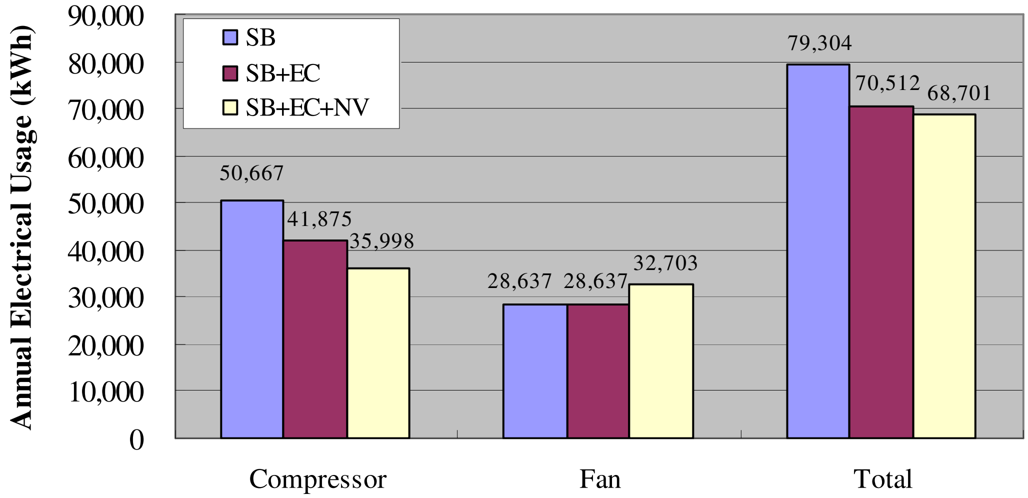


Figure 28: Simulated electrical energy usage for different ventilation strategies (class school wing in California Climate Zone 10).

6.2 Comparisons with Energy-10

Energy-10 is a conceptual design tool for low-energy buildings developed under sponsorship of the U.S. Department of Energy (DOE). The program performs transient, hour-by-hour load calculations for small commercial and residential buildings and allows comparisons of different energy savings strategies. The underlying methods used in Energy-10 are very similar to those used in VSAT. However, VSAT focuses on energy savings due to different ventilation strategies, whereas Energy-10 considers more conventional design changes such as day-lighting, air leakage control, glazing, shading, economizer, thermal mass, passive solar heating and high efficiency equipment. More information on Energy-10 can be found at www.sbicouncil.org or from the user manual.

Since Energy-10 is an accepted tool for analysis of small commercial building, it was chosen for benchmarking predictions of VSAT for a base case system with a night setup/setback thermostat and no economizer. The office-building prototype was chosen for this case study and comparisons of monthly equipment loads and energy consumptions were performed in two locations, Madison, WI, and Atlanta, GA.

There are some basic differences in the modeling approaches in Energy-10 and VSAT that had to be considered. VSAT neglects the effects of cycling on furnace efficiency, whereas Energy-10 includes a significant penalty for cycling. For the purposes of comparison, the part-load effects of furnace cycling were not included in the Energy-10 results. However, part-load effects for the air conditioning equipment were included for both models.

A window in Energy-10 is characterized with a rough-frame opening dimension(the hole left by the framers), a glazing type, and a frame type. The U-value is calculated from the dimensions and the U-values of the glass and frame. In VSAT, the window U-value is simply a given value in the building description and no frame is assumed. Solar transmittances and shading coefficients are also inputs in VSAT, whereas these values are calculated from a windows library in Energy-10. For comparison purposes, a window assembly was built in Energy-10 that had an effective U-value and transmittance very similar to that in VSAT.

Energy-10 weather files are constructed using the 1994 and 1995 updated TMY2 weather files. This update is based on 30 years of data, rather than 20 years, and incorporates new and improved solar radiation information from the 1992 National Solar Radiation Data Base. The weather data in VSAT used for comparison purposes is TMY data.

The prototypical office was modeled in both Madison and Atlanta. The air-conditioner and furnace equipment models assumed a rated EER of 11 and efficiency of 85%, respectively. In VSAT, the supply fan power was assumed to be 0.5 W/cfm. This value corresponds to a fan efficiency of 11.78% and 0.5 inches H_2O system static pressure as entered in Energy-10. Infiltration was neglected for both models. The occupied zone set point for cooling was 23.89°C with a night setup to 29.44°C. The occupied zone set point for heating was 21.11°C with a night setback to 15.56°C.

Table 21 gives equipment sizing determined by VSAT for Madison and Atlanta. These equipment sizes were specified in Energy-10.

Table 21: Equipment Sizing Results from VSAT

	Office - Madison, WI	Office - Atlanta, GA
AC Rated Total Cap., Btu/h	210180	210260
AC Rated Sens. Cap., Btu/h	153064	162822
Furnace Rated Cap., Btu/h	252924	148583
Total Air Flow, cfm	6130	6133
Ventilation Air Flow, cfm	924	924
Office Floor Area, ft^2	6600	6600

Figures 29 – 31 give monthly electricity for the condensing units (compressors and condenser fans), furnace gas input, and supply fan power for both VSAT and Energy-10 in Atlanta, GA. Figures 32 – 34 give similar results for Madison, WI. The trends and absolute magnitudes are very similar for predictions obtained with VSAT and Energy-10. In general, VSAT tends to give slightly higher condensing unit energy and lower gas input energy than Energy-10. Tables 22 and 23 gives tabulated results along with percentage differences between Energy-10 and VSAT for Atlanta and Madison.

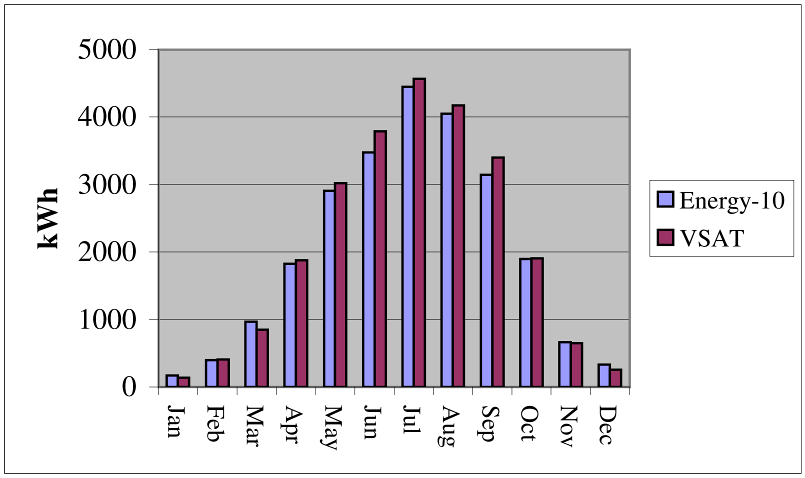


Figure 29. Monthly Electrical Consumption for Cooling – Atlanta, GA

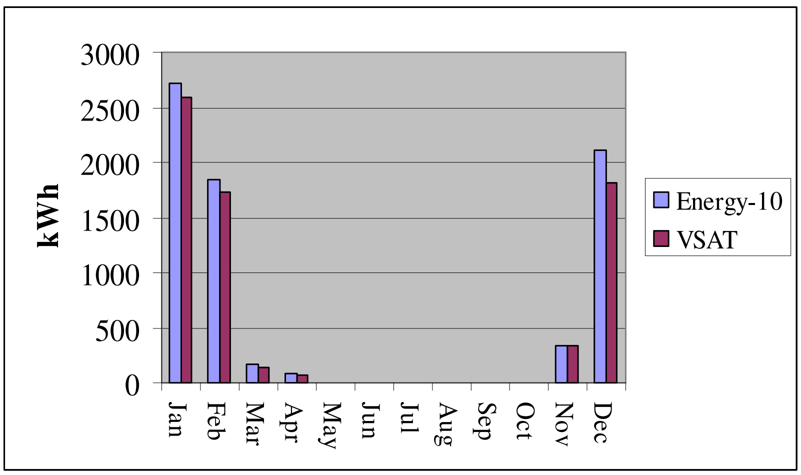


Figure 30. Monthly Furnace Gas Input – Atlanta, GA

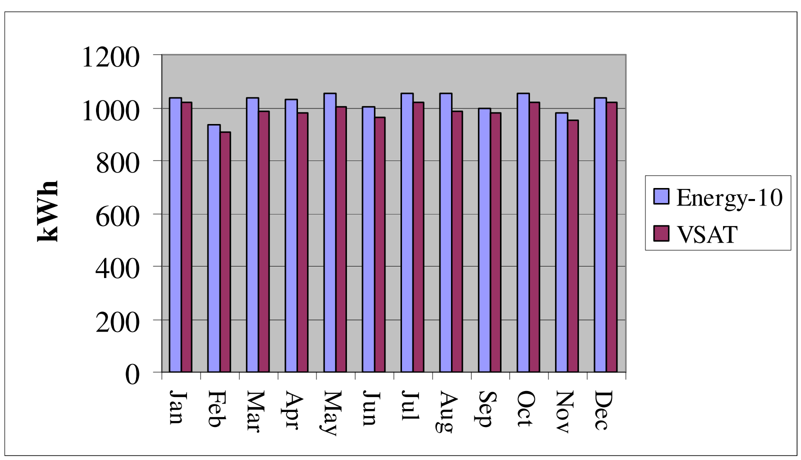


Figure 31. Monthly Supply Fan Power Consumption – Atlanta, GA

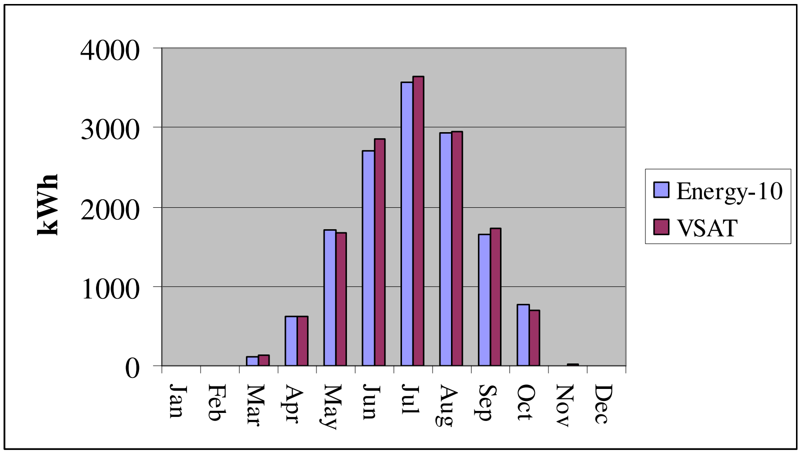


Figure 32. Monthly Electrical Consumption for Cooling –Madison, WI

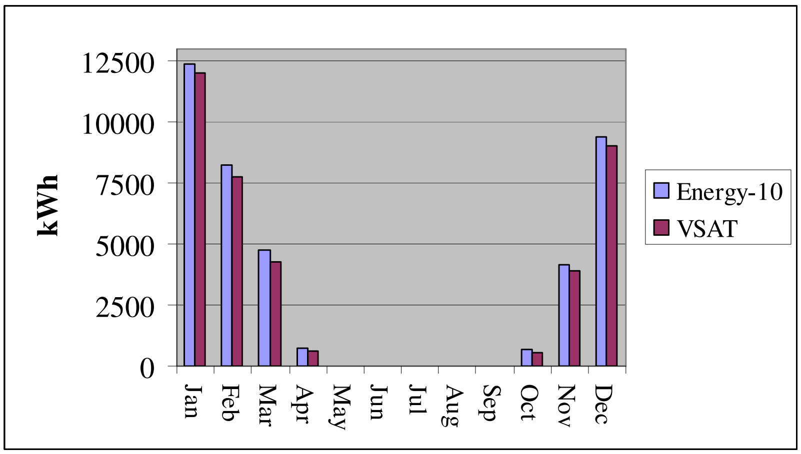


Figure 33. Monthly Furnace Gas Input – Madison, WI

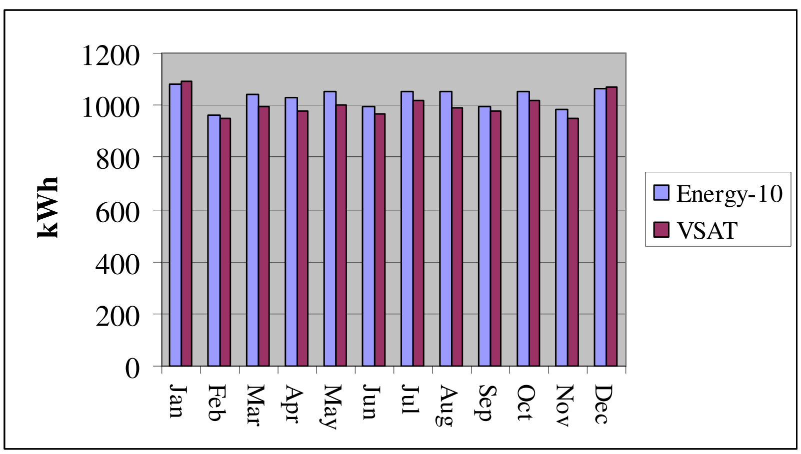


Figure 34. Monthly Supply Fan Power Consumption – Madison, WI

Table 22: VSAT and Energy-10 Results – Atlanta, GA

	AC kWhr		Furnace kWhr		Fan kWhr	
Month	Energy-10	VSAT	Energy-10	VSAT	Energy-10	VSAT
Jan	168	139	2712	2589	1038	1020
Feb	397	407	1851	1738	937	909
Mar	968	849	171	142	1037	987
Apr	1828	1878	78	74	1031	981
May	2905	3021	0	0	1053	1003
Jun	3474	3790	0	0	1000	966
Jul	4448	4570	0	0	1054	1018
Aug	4050	4173	0	0	1054	987
Sep	3144	3398	0	0	999	981
Oct	1898	1905	0	0	1053	1018
Nov	662	650	332	345	982	951
Dec	330	258	2118	1815	1037	1018
Yr	24272	25038	7262	6703	12275	11839
	AC error	3.06%	Furnace error	-8.35%	Fan error	-3.68%

Table 23: VSAT and Energy-10 Results – Madison, WI

	AC kWhr		Furnace kWhr		Fan kWhr	
Month	Energy-10	VSAT	Energy-10	VSAT	Energy-10	VSAT
Jan	0	0	12378	12018	1082	1092
Feb	5	0	8247	7779	960	948
Mar	110	128	4779	4264	1039	994
Apr	622	618	711	622	1030	981
May	1716	1664	11	12	1053	1002
Jun	2705	2852	0	0	998	966
Jul	3561	3636	0	0	1054	1018
Aug	2934	2943	0	0	1053	987
Sep	1650	1728	0	18	998	981
Oct	766	687	694	523	1053	1018
Nov	2	20	4178	3927	982	951
Dec	0	0	9414	9023	1065	1070
Yr	14070	14276	40412	38186	12367	12008
	AC error	1.44%	Furnace error	-5.83%	Fan error	-2.99%

SECTION 7: REFERENCES

ASHRAE. 2000. *ASHRAE Handbook: HVAC Systems and Equipment*, Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1999. "ANSI/ASHRAE Standard 62-1999, "Ventilation for Acceptable Indoor Air Quality", Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1993. *ASHRAE Handbook: Fundamentals*, Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Brandemuehl, M.J. and Braun, J.E., "The Impact of Demand-Controlled and Economizer Ventilation Strategies on Energy Use in Buildings, ASHRAE Transactions, Vol. 105, Pt. 2, pp. 39-50, 1999.

Brandemuehl, M.J., Gabel, S., and Andresen, I. 1993. *HVAC2 Toolkit: Algorithms and Subroutines for Secondary HVAC System Energy Calculations*. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.

Braun, J.E. and Brandemuehl, M.J. 2002. *Savings Estimator Technical Reference Manual.*

Carrier Corporation. 1999. "Product Data: 62AQ Energy$Recycler Energy Recovery Accessory For 3 to 12.5 Ton Rooftops", Form 62AQ-1PD. Syracuse, New York.

Carnes Company. 1989. "Energy Recovery Wheel Design Manual," Verona, WI.

Chaturvedi, N. and Braun, J.E. "An Inverse Gray-Box Model for Transient Building Load Prediction," International Journal of Heating, Ventilating, Air-Conditioning and Refrigeration Research, Vol. 8, No. 1, pp. 73-100, 2002.

Duffie, J.A. and Beckman, W.A., Solar Engineering of Thermal Processes, Wiley—Interscience, 1980.

DOE-2. 1982. Engineer's Manual, Version 2.1A. Lawrence Berkeley Laboratory and Los Alamos National Laboratory.

Huang, Y.J., Akbari, H., Rainer, L., and Ritschard, R.L. 1990. *481 Prototypical Commercial Buildings for Twenty Urban Market Areas (Technical documentation of building loads data base developed for the GRI Cogeneration Market Assessment Model).* LBL Report 29798, Lawrence Berkley National Laboratory, Berkley CA.

Huang, Y.J., and E. Franconi. 1995. *Commercial Heating and Cooling Loads Component Analysis.* LBNL Report 38970, Lawrence Berkeley National Laboratory, Berkeley, CA.

Kays, W.M., and M.E. Crawford. 1980. *Convective Heat and Mass Transfer.* New York: McGraw-Hill.

Klein, H., Klein, S.A., and Mitchell, J.W., 1990, “Analysis of regenerative enthalpy exchangers,” International *Journal of Heat and Mass Transfer*, Vol. 33, No.4, pp. 735-744.

Maclaine-cross, I.L. 1974. *A Theory of Combined Heat and Mass Transfer in Regenerators.* Ph.D. Thesis in Mechanical Engineering, Monash University, Australia.

Seem, J.E., Klein, S.A., Beckman, W.A., and Mitchell, J.W., 1989, “Transfer functions for efficient calculations of multi dimensional heat transfer,” *Journal of Heat Transfer-Transactions of the ASME*, Vol. 111, No.1, pp. 5-12.

Stiesch, G., S.A. Klein, J.W. Mitchell. 1995. *Performance of Rotary Heat and Mass Exchangers.* HVAC&R Research 1(4): 308-324.

TRNSYS. 2000. *TRNSYS Users Guide: A Transient Simulation Program.* Solar Energy Laboratory, University of Wisconsin-Madison.

INITIAL COOLING AND HEATING SEASON FIELD EVALUATIONS FOR DEMAND-CONTROLLED VENTILATION

Submitted to

California Energy Commission

As Deliverables 3.1.3a and 3.1.4a

Prepared by

Tom Lawrence and James E. Braun

Purdue University

February 2003

Table of Contents

1. Introduction

This report summarizes analyses of initial cooling and heating season data from California field test sites for evaluation of demand controlled ventilation (DCV). Deliverable 3.1.1a (2001) provided a description of the field sites and data collection hardware and procedures. The current report provides estimates of energy savings and CO_2 concentrations associated with application of DCV at these sites. The baseline of comparison is fixed ventilation rates based upon ASHRAE Standard 62-1999 (ASHRAE, 1999). A separate report (Deliverable 4.2.3a) will provide simulation evaluations of DCV and comparisons with other competing technologies, including enthalpy exchangers and heat recovery heat pumps.

Each of the sites has a differential enthalpy economizer and is capable of operating with and without DCV. The field sites were chosen to accommodate side-by-side comparisons between fixed ventilation and DCV for different building types and climates. The strategies were rotated at each location to allow both both side-by-side and same-store comparisons.

1.1 Field-Site Descriptions

Field sites have been established for three different building types in two different climate zones within California. The building types are: 1) McDonalds PlayPlace® areas, 2) modular school rooms, and 3) Walgreens drug stores. In each case, nearly duplicate test buildings were identified in both coastal and inland climate areas.

The PlayPlace areas are isolated from the main dining area and have separate packaged rooftop HVAC unit(s). Heating is provided by natural gas burners. Two restaurants sites are located approximately 15 miles apart in the San Francisco Bay area (south of Oakland and north of San Jose). Two other restaurant sites are in the Sacramento area.

The modular schoolrooms are typical of thousands employed throughout California and the United States. They use a single sidewall mounted packaged heat pump system. Two schoolrooms are located in Oakland and two are in Woodland, just east of Sacramento.

The drug stores selected for this study are larger than the other field sites and use five rooftop units that service the store and pharmacy areas. Due to the larger number of HVAC units at the Walgreens sites, only one store in each climate type is being monitored. One store is near Riverside and the other is in Anaheim.

The two alternative control strategies compared are DCV with economizer control (DCV On) and economizer cooling only (DCV Off). With the DCV On strategy, the return air CO_2 setpoint was 800 ppm_v. When the return air CO_2 concentration is below the setpoint, the outdoor air ventilation damper is fully closed. Otherwise, the Honeywell controller provides feedback control of the damper position. For the DCV Off mode, a minimum damper position is set so as to provide the required outdoor airflow as specified in ASHRAE Standard 62-1999 (ASHRAE, 1999). The fixed damper position that satisfies the standard was estimated to be 40% for the McDonalds and the modular schools and 20% open for the Walgreens stores. However, field airflow measurements at one McDonalds store indicates that the actual total supply airflow varies significantly with damper position. This impacts the actual amount of ventilation air provided, as discussed later in the report.

Installation at the field test sites began in late 2000 with installation, checkout and debugging finished by the end of 2001 for the McDonalds and modular schoolroom sites. The Walgreens store installation and debugging continued into 2002. At this time, field site results are only available for the McDonalds PlayPlace and modular schoolroom sites, since the Walgreens store control and monitoring equipment only became fully operational in late summer of 2002. A detailed description of the field test sites and the data collection system is included in Deliverable 3.1.1a (2001).

1.2 Data Collection Methodology

The field measurements for HVAC equipment include electric power, integrated electrical energy, digital control signals for the gas valve and supply fan, ambient, return, and mixed air temperature and humidity, supply air temperature, and return air CO_2 concentration.

The power is calculated from voltage and current readings for each unit (fans plus compressor). For the Bradshaw Road and Milpitas sites that have two rooftop units, only direct power measurements from one of the units are available, but they are duplicate systems. Operation of the second rooftop unit is monitored via the digital control signals indicating fan, cooling or heating being on. Since the modular school sites use a single phase electrical power connection, separate monitoring of the total unit and compressor power is performed.

Data are collected every five minutes using Virtual Mechanic monitoring equipment. Data are downloaded to the Field Diagnostic Services (FDS) main server on a daily basis using a cell phone. An FDS website provides direct access to the data. A screening analysis program is used to check for erroneous data and compute hourly averages.

For the rooftop units at the McDonalds sites, direct measurement of gas consumption is not available. For a given period of time, the natural gas consumption is estimated as

$$\text{Gas usage (ft}^3\text{)} = \frac{\text{Heater On Time (hrs)} \times \text{Unit Heater Rating (Btu/hr)}}{1000\ \text{Btu / ft}^3\,\text{natural gas}}$$

where the on time is estimated from the gas valve control signal and the input rating is from the manufacturer's nameplate data.

Heater input ratings for the York units at the McDonalds PlayPlace areas are:

- Milpitas: 125,000 Btu/hr (each unit, total 2 units)
- Castro Valley: 204,000 Btu/hr (1 unit)
- Bradshaw Road: 100,000 Btu/hr (each unit, total 2 units)
- Watt Avenue: 200,000 Btu/hr (1 unit)

The modular school and retail store field sites use heat pump rooftop units and therefore the energy required for heating is determined directly from the power measurements.

For each site, hourly local weather data were also collected at National Weather Service (NWS) stations closest to each test site. Table 1 shows the corresponding NWS stations for each test site.

Table 1 – National Weather Service Stations for the Test Sites

Test Site	NWS Station ID	Station Description	Approximate Distance to Test Site
Milpitas	KSJC	San Jose airport	6 miles
Castro Valley	KHWD	Hayward airport	5 miles
Oakland school	KOAK	Oakland airport	7 miles
Woodland school	KSMF	Sacramento airport	10 miles
Bradshaw Road	KSMF	Sacramento airport	12 miles
Watt Avenue	KSMF	Sacramento airport	9 miles

1.3 Methodologies for Comparing Strategies

This report provides comparisons of energy usage at the field sites for the systems controlled both with and without DCV in both cooling and heating. Various methods were developed to compare the energy usage and are summarized as follows:

1. Direct side-by-side comparisons – Nearly identical sites were chosen in the northern California climates to allow direct side-by-side comparisons for the same time periods. As a check on the differences between sites, it is important to also compare energy use with both sides operating in the same mode (e.g., DCV On or DCV Off).
2. Correlated daily energy usage – This approach involves comparison of average daily energy use for heating or cooling at the same site. Total daily energy usage is correlated as a function of average ambient temperature for different time periods when the DCV is on and off. Separate correlations are developed for DCV On and Off and then used to compare energy use for a given daily ambient condition or over a period of time (e.g., cooling or heating season).
3. Correlated hourly energy usage – This is an improvement on approach #2, whereby hourly energy usage is correlated using a time-series model with many inputs, including ambient temperature and dew point, zone temperature, estimated occupancy, HVAC system status (occupied or unoccupied), and solar radiation levels (estimated or "clear sky" approximations). Current and

previous inputs and previous predicted energy usages are included as inputs to model. Separate models are developed for DCV On and Off. Training requires a non-linear optimization to identify parameters that minimize the errors between model predictions and measurements. The trained models are then used with actual or standardized weather data, occupancy patterns, etc. to compare cooling or heating season energy usage for each site both with and without DCV control.

4. Calibrated VSAT simulations – VSAT simulations have been prepared for the field sites in northern California and provide hourly predictions of energy usage. In order to reduce errors between model predictions and measurements, some of the least well-known parameters in VSAT (e.g, window shading coefficients) are tuned using measured data. The result is a "calibrated" model that can be used to predict cooling and heating season energy use with both DCV On or Off.

Method 1 involves direct comparisons of the field data for different sites, whereas methods 2, 3, and 4 allow "virtual" side-by-side comparisons of DCV On and DCV Off strategies for the same time period and same site. The direct comparison approach can be convincing, but is only appropriate if the two sites have nearly identical weather data, internal gains, and occupancy profiles. Furthermore, the comparisons are limited to the time periods over which data were collected. The modeling approaches (methods 2 – 4) allow interpolation and extrapolation of performance for different weather conditions than were encountered during testing. The models can be "driven" with weather data to yield seasonal comparisons of cooling and heating energy usage. Method 2 only considers the effect of average ambient conditions and is the least accurate of the modeling approaches considered. Method 3 is a much more detailed modeling approach, but requires significantly more data and effort. The typical approach is to develop a model using a training data set and then test the model with a separate test data set. However, due to the limited amount of data available for most cases, the models outlined in this report were trained using the complete set of data available. Method 4 has the advantage of allowing the models to be tuned for one mode of operation in a single season, but used to predict energy usage in both seasons with both DCV On and Off. The use of calibrated simulation models (Method 4) is described in the literature (Schuldt and Romberger, 1998) and in performance measurement and verification protocols (DOE, 1997).

Evaluation of the impact of DCV control on the indoor air quality was performed by comparing the CO_2 concentrations in the return air for both DCV On and DCV Off during hours of potential occupancy.

2. Energy Savings for Cooling

2.2 McDonalds PlayPlace Areas

Side-by-Side Comparisons

A side-by-side comparison of the McDonalds data was only possible with the Bay Area McDonalds since one of the Sacramento area sites (Watt Avenue) had a problem with the damper control.

During the month of August and early September, variations in the DCV control settings were made at the Milpitas and Castro Valley sites to allow side-by-side comparisons. A time period was also run with both sites set to DCV Off to compare inherent energy usage patterns. Figure 1 shows daily energy usage for cooling from August 29 to September 8 with DCV Off for both sites. The Castro Valley site had slightly higher energy consumption (82.8 kW-hr per day) compared to the Milpitas site (80.0 kW-hr per day), a difference of about 3.5%.

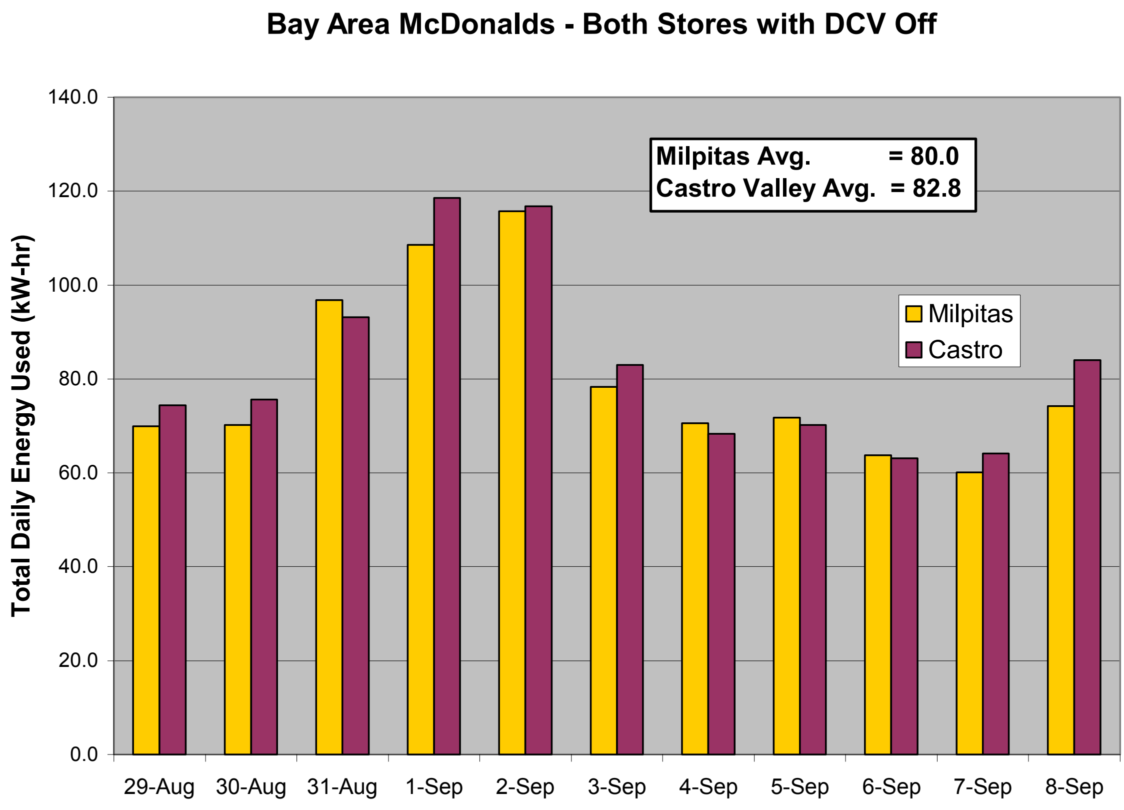


Figure 1. Comparison of Energy Use for Cooling at Bay Area McDonalds with DCV Off

Figure 2 shows side-by-side comparisons of daily cooling energy usage for DCV On and DCV Off at the two sites during the period August 9 - 28. The strategies were alternated between the two sites, but the savings for DCV On were nearly the same regardless of which sites were on and off. Average measured daily energy savings for DCV On was about 14% for this time period.

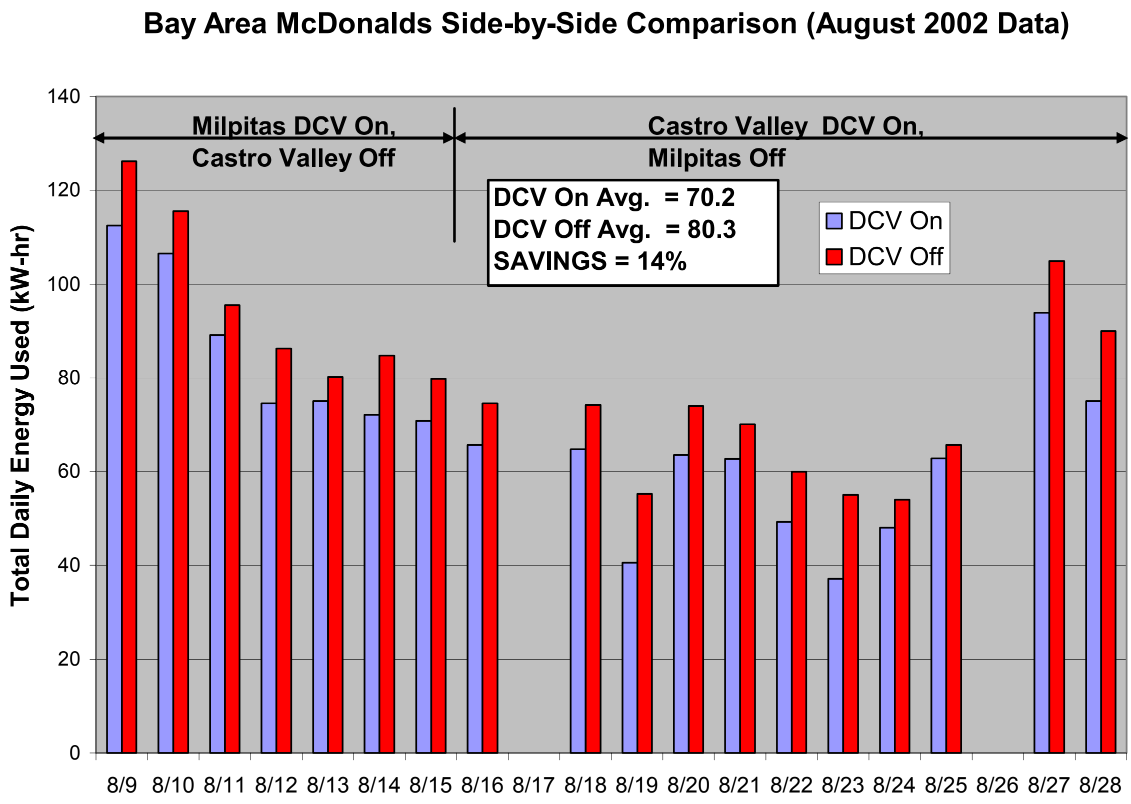


Figure 2. Comparison of Cooling Energy Use for DCV On and Off at Bay Area McDonalds

Correlated Daily Energy Usage

Figure 3 shows daily energy usage for cooling as a function of daily average ambient temperature for the Milpitas site (Bay area) for both DCV On and Off. The daily data correlates relatively well as a linear function of ambient temperature. For a hot day with an average temperature of 80º F, the estimated savings are about 12%. Figure 4 shows similar results for the other Bay area site (Castro Valley). In this case, the savings are a little smaller than for the Milpitas site. This may be because this site has a greater occupancy, leading to higher ventilation rates for DCV On as compared with Milpitas.

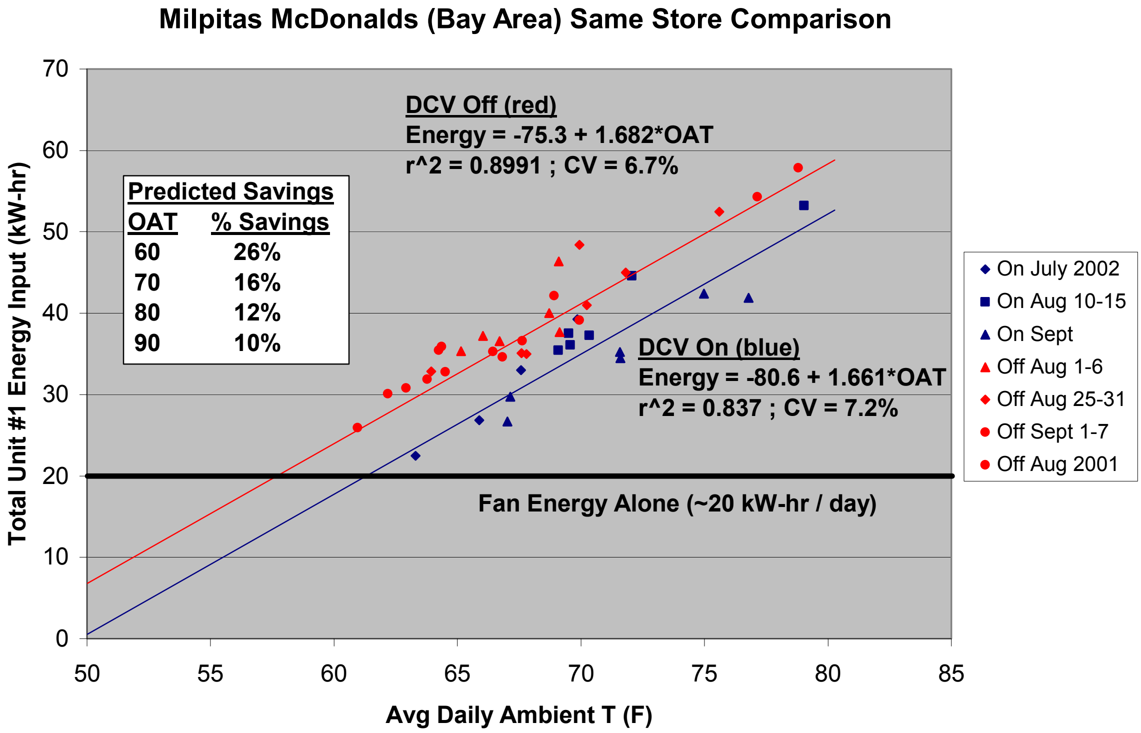


Figure 3. Correlated Daily Cooling Energy Use for DCV On and Off at Milpitas (Bay Area) McDonalds Site

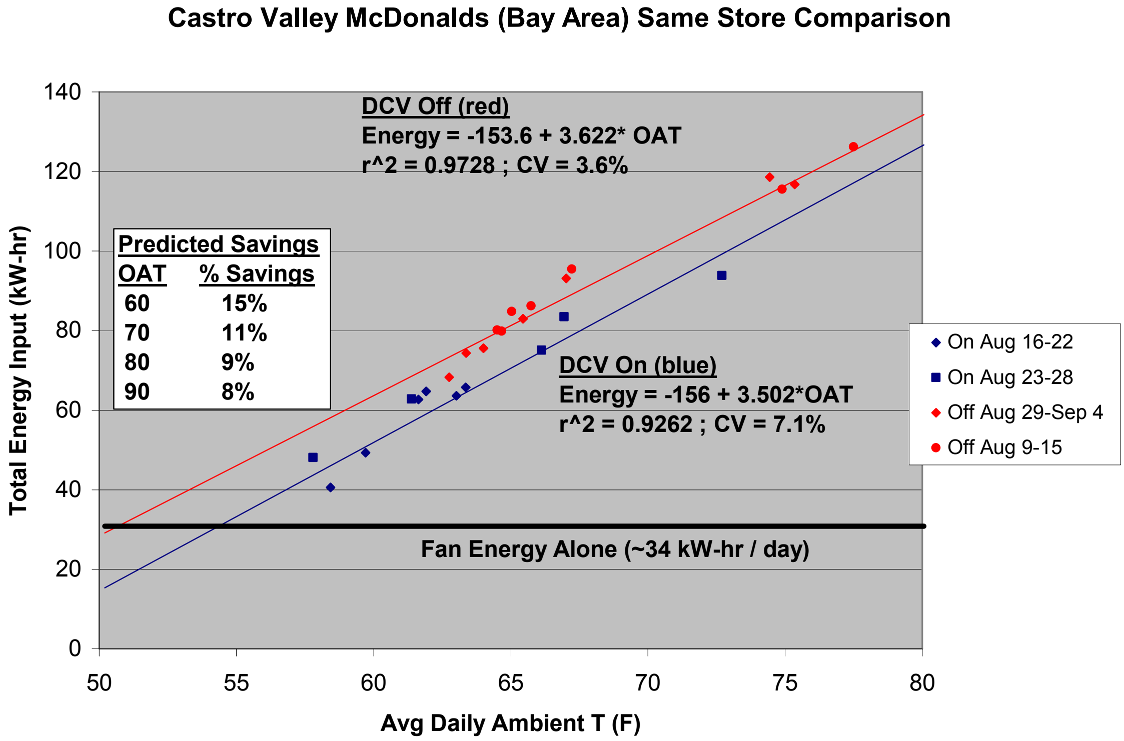


Figure 4. Correlated Daily Cooling Energy Use for DCV On and Off at Castro Valley (Bay Area) McDonalds Site

Figure 5 shows daily energy usage for cooling as a function of daily average ambient temperature for the Bradshaw (Sacramento area) McDonalds for DCV On and Off. For a hot day with an average temperature of 80º F, the estimated savings are about 28%. These savings are considerably larger than those for the Bay area sites. For the same average daily temperature, the daytime temperatures are higher for Sacramento than the bay area leading to larger ventilation loads and greater savings with DCV. Also, the occupancy at the Bradshaw site appears to be lower than for the other McDonalds sites.

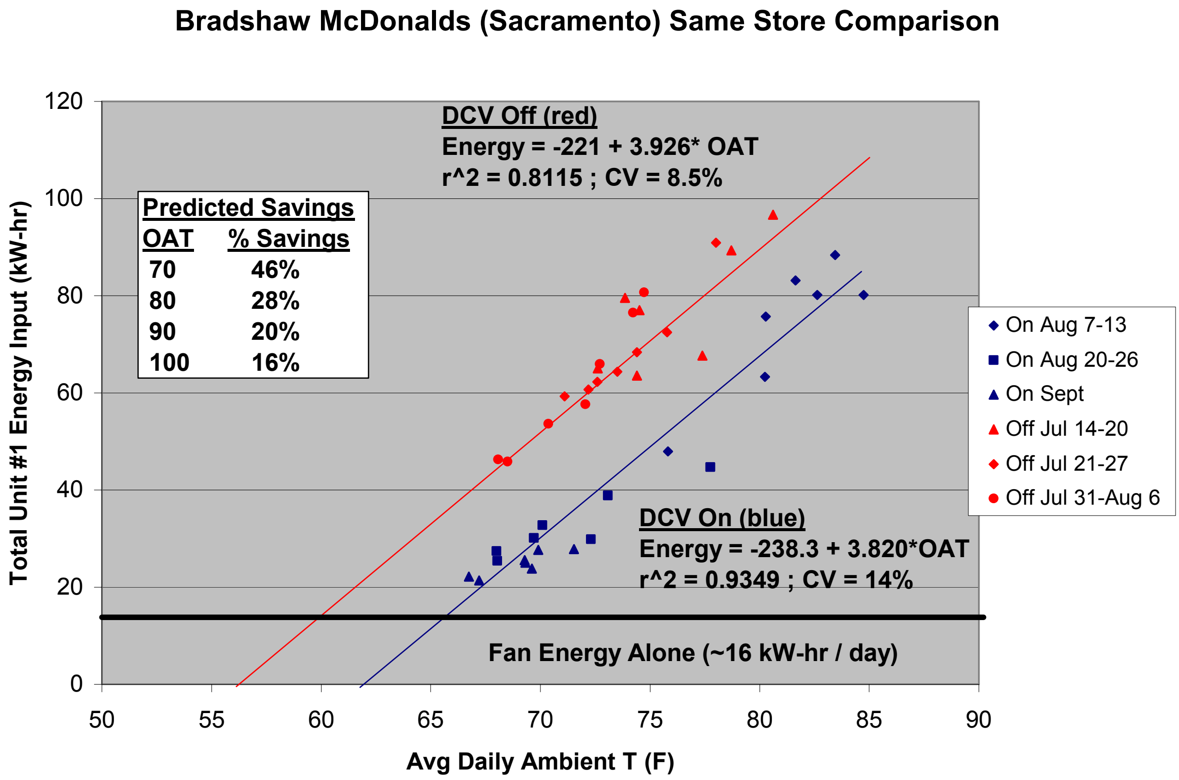


Figure 5. Correlated Daily Cooling Energy Use for DCV On and Off at Bradshaw (Sacramento) McDonalds Site

It was not possible to perform comparisons for the Watt Avenue McDonalds in Sacremento because the outdoor air damper control and position feedback indicator do not appear to be working properly

Table 2 summarizes the energy savings versus daily average ambient air temperature for the three McDonalds sites.

Table 2. Measured Savings Percentages with DCV On Control Strategy at McDonalds PlayPlace Areas

Daily Average Temperature (F)	Bradshaw Road (Sacramento)	Milpitas (Bay Area)	Castro Valley (Bay Area)
60	Not applicable (fan only)	26%	15%
70	46%	16%	11%
80	28%	12%	9%
90	20%	10%	8%

Correlated Hourly Energy Usage

Separate "blackbox" models were developed for DCV On and DCV Off at the Bradshaw, Castro Valley and Milpitas McDonalds sites. The Watt site data was not evaluated due to the damper control problem. The available model training data sets for the "on" and "off" cases ranged from around two weeks to over 60 days, depending on the site and control strategy. The models predict hourly condenser unit energy usage, which includes energy for both the compressor and condenser fan. The supply fan energy is added to the predictions of condenser unit energy. The installed field monitoring system measures the total rooftop unit power input, which includes energy for the compressor(s) and supply and condenser fans. However, it is straightforward to determine the supply fan power consumption during times when the compressor and condenser fan are off. The supply fan power is relatively constant. Figure 6 shows sample hourly profiles for air conditioning energy usage for a typical day at a McDonalds site. The supply fan energy is a significant fraction of the total energy consumption for air conditioning.

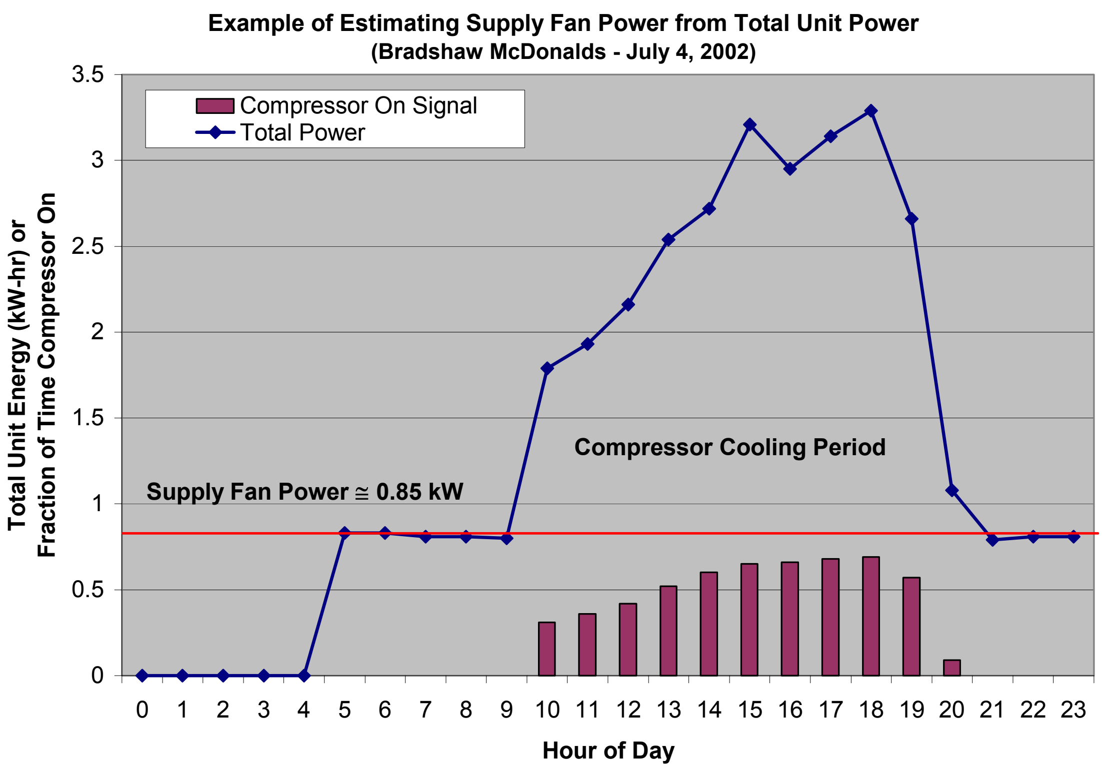


Figure 6. Example of Estimating Supply Fan Power at McDonalds Sites

A number of different model formats were evaluated. The model predictions were compared to the actual measured data using several metrics, including total energy usage, hourly root mean square error and overall peak hourly energy demand. Table 3 summarizes the performance of the "best" models developed for each site. In general, the models do a very good job of predicting hourly and total energy usage over the training period. The predicted power input to the condensing unit is added to a constant supply fan power to give the total packaged HVAC unit energy consumption when doing control strategy comparisons.

Table 3. Blackbox Model Performance Compared to Actual Measured Data During Cooling Season Training Periods for McDonalds PlayPlace Areas

Performance Metric	Control Strategy	Bradshaw Road	Milpitas	Castro Valley
Total energy usage	DCV On DCV Off	+0.4% -2.3%	-4.0% +0.1%	-4.6% -2.3%
Root mean square error (hourly kW-hr)	DCV On DCV Off	0.55 0.41	0.40 0.37	0.39 0.61
Peak cooling energy demand	DCV On DCV Off	+1.2% +0.1%	-16% -6%	+0.3% -0.1%

Time period for model training in 2002:
Bradshaw DCV On: Aug 7-14, 20-27, 30. Sept 1,4-9 and 13-30.
DCV Off: Mar 27-Apr 9, April 20-26, May 2-7, July 14-Aug 5
Milpitas DCV On: April 24-30, July 13-15, 18-19, 31, Aug 7-10, 12-15,
Sept 9-10, 17-19, Oct 2-7
DCV Off: Aug 14 – Sept 7

Figures 7 and 8 show sample comparisons between hourly predictions and measurements of compressor energy usage for the Bradshaw site. Figure 7 shows a week when the system was operating in DCV Off mode, whereas Figure 8 is a week of DCV On operation. The models capture the general time variations of the energy usage for the particular control strategy in place at the site during the given timeframe. For any given hour, the models can underpredict or overpredict the requirements due to uncertainties in occupancy and other unmeasured driving conditions. However, on average, the models do extremely well in predicting the behavior.

Figures 7 and 8 also show "virtual" side-by-side comparisons between DCV On and Off for the Bradshaw site. For these two particular weeks combined, the estimated savings in energy for DCV On is about 45% for the condensing unit energy usage alone or 30% when considering the total rooftop unit energy consumption (condensing unit + supply fan). These values correspond closely to the same store data comparison trends in Figure 5, where the predicted total unit energy savings based on the regression lines is around 31% for the same two weeks combined.

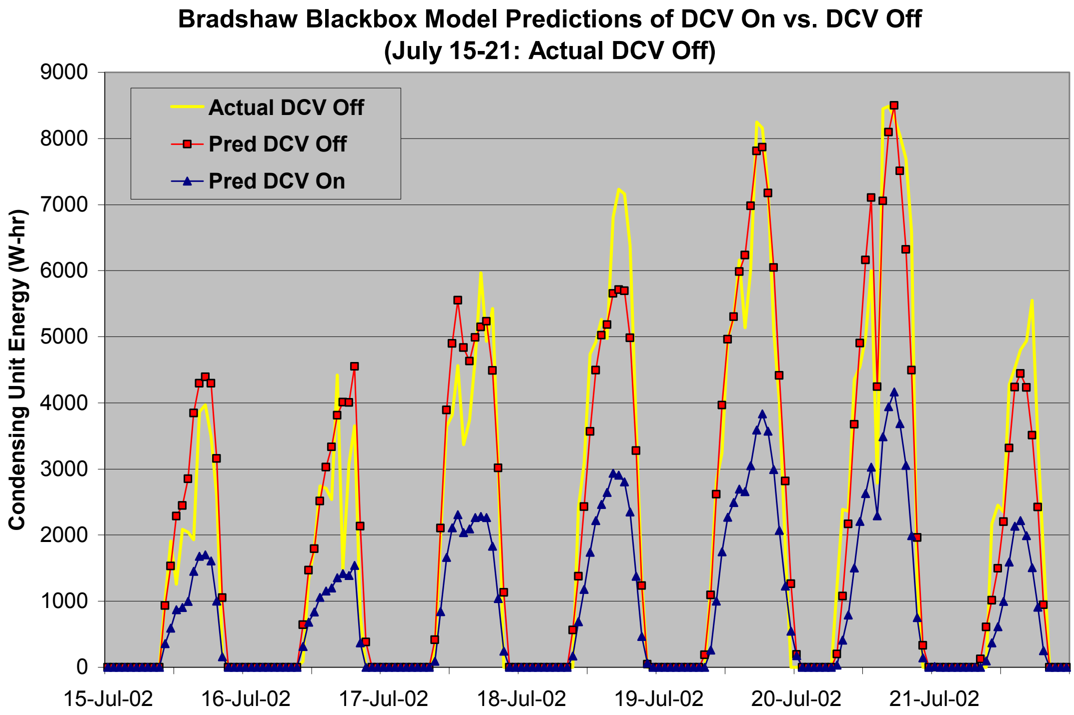


Figure 7. Hourly Blackbox Model Predictions of Compressor Energy for DCV On and Off at Bradshaw (Sacramento) McDonalds Site for Period with DCV Off

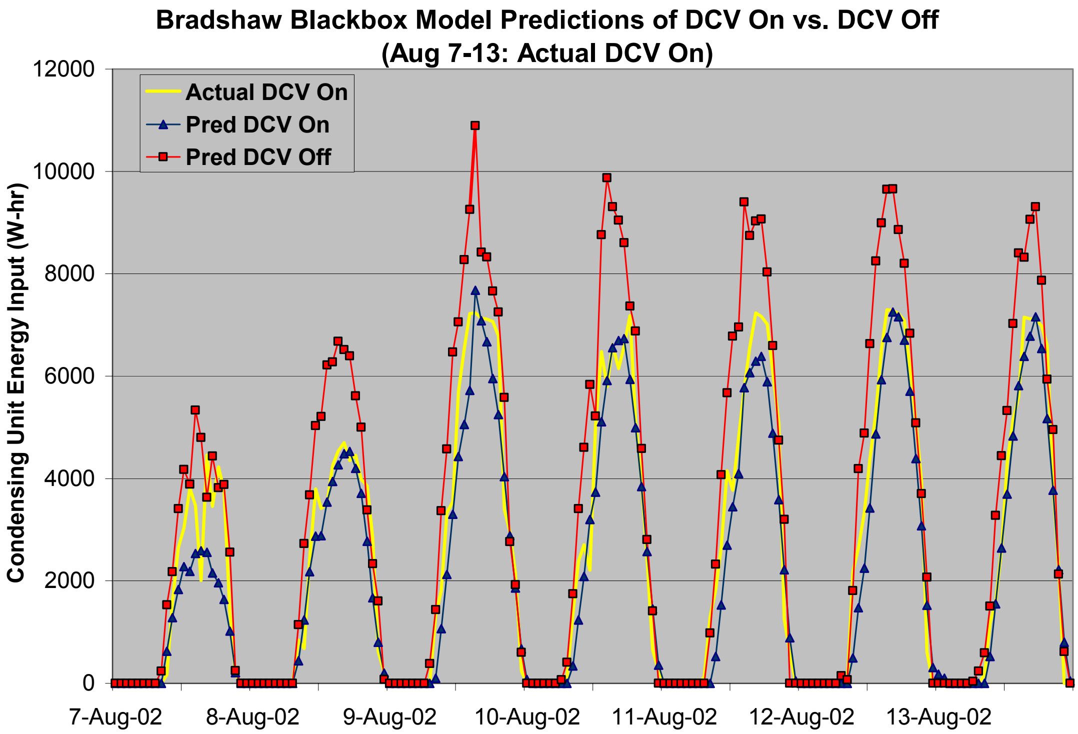


Figure 8. Hourly Blackbox Model Predictions of Compressor Energy for DCV On and Off at Bradshaw (Sacramento) McDonalds Site for Period with DCV On

Since the models are nonlinear, they may not extrapolate well to conditions far outside of those encountered during training. Since a wide range of data was available for the Bradshaw and Milpitas sites, it is reasonable to use these models to predict annual energy savings. However, only twelve days of data in August were available for the Castro Valley site and therefore the blackbox models were not used to estimate seasonal savings at this location.

Table 4 gives comparisons of annual cooling system performance for DCV On and DCV Off at the Bradshaw and Milpitas McDonalds PlacePlaces. The models were then applied to the entire year for cooling savings estimates. Significant energy savings are observed for DCV at the Bradshaw site in Sacramento and smaller savings. The savings in condensing unit energy were 35% and 16% for the Bradshaw and Milpitas sites, respectively. The total annual air conditioning cost savings were smaller (23% and 6%, respectively) because the supply fans operates continuously during occupied times for both strategies and fan energy is a significant fraction of the total energy usage.

The estimates in Table 4 were obtained using the blackbox models for the entire year using 2002 weather data and estimates of occupancy obtained using CO_2 measurements. For a few days, occupancy information was not available and average hourly occupancy levels determined from measurements for the other days were used. Different average occupancy patterns were determined from the measurements for weekends and weekdays. The system comparisons include system supply fan and condensing unit (compressor plus condenser fan energy) for DCV On and Off, condensing unit savings, total unit energy savings, percent energy savings, and percent reduction in peak power for DCV On as compared with DCV Off. The supply fan energy was estimated using average values observed from the field data, as was illustrated earlier in Figure 6.

A note is needed regarding the setting for the ventilation airflow. The ventilation flow used for the DCV On scenario is for the outdoor air damper to be closed until the CO_2 readings reach the setpoint, and then the damper modulates as needed. The original intent for the DCV Off test case was to set up the ventilation flow to be a constant value equal to that prescribed by ASHRAE Standard 62-1999. The ventilation flow is set by the outdoor air damper position. Based on the manufacturer's rating for the supply fans for the various rooftop units and a rough analysis of the entire airflow system, the setpoint position for the damper was determined. The setpoints implemented into the test program are 40% damper for the McDonalds and modular schools. Field measurements were made later in the test program when time allowed. These measurements indicate that the actual supply flow rate varies with damper position (as the ratio between outdoor air and return air cahnges) and is generally higher than the manufacturer's rating Figure 9 shows total supply air flow rate and ventilation flow versus outdoor damper position for the Watt Avenue McDonalds in Sacramento (similar to the systems at the other sites). As the outdoor damper opens, the supply air flow increases as the overall pressure drop is reduced. There is a peak in the supply air flow at around 60% damper position, and this flow is significantly (around 45%) higher than the manufacturer's rating. At the 40% damper setpoint, the total supply flow is around 5500 cfm and the ventilation flow is around 2700 cfm. The original design intent was for 1600 cfm ventilation flow, corresponding to an occupancy of 80 people and 20 cfm per person. Thus, the net result is a higher than anticipated ventilation air flow rate for the McDonalds sites.

The measured flow for the modular schoolrooms indicated a close match between desired airflow and the actual measured values at a 40% damper position. The measured ventilation air was around 450 cfm at the 40% damper position, which is the same as the test plan design of 15 cfm per person with a 30 person occupancy.

Table 4. Full Year Cooling Energy Savings Predicted Using Blackbox Models for McDonalds PlayPlace Areas

	Bradshaw	Milpitas
Fan energy, DCV On (kWh)	7,604	8,150
Fan energy, DCV Off (kWh)	7,604	8,150
Condensing unit, DCV On (kWh)	9,536	3,768
Condensing unit, DCV Off (kWh)	14,695	4,486
Total savings, condensing unit only (kWh)	5,159	718
Total savings, condensing unit + fan energy	5,159	718
% annual electrical energy savings, condensing unit alone	35.1%	16.0%
% annual electrical energy savings, condensing unit + fan	23.2%	5.6%
Savings in maximum hourly peak demand (peak DCV Off – peak DCV On, kW)	5.7	-0.4

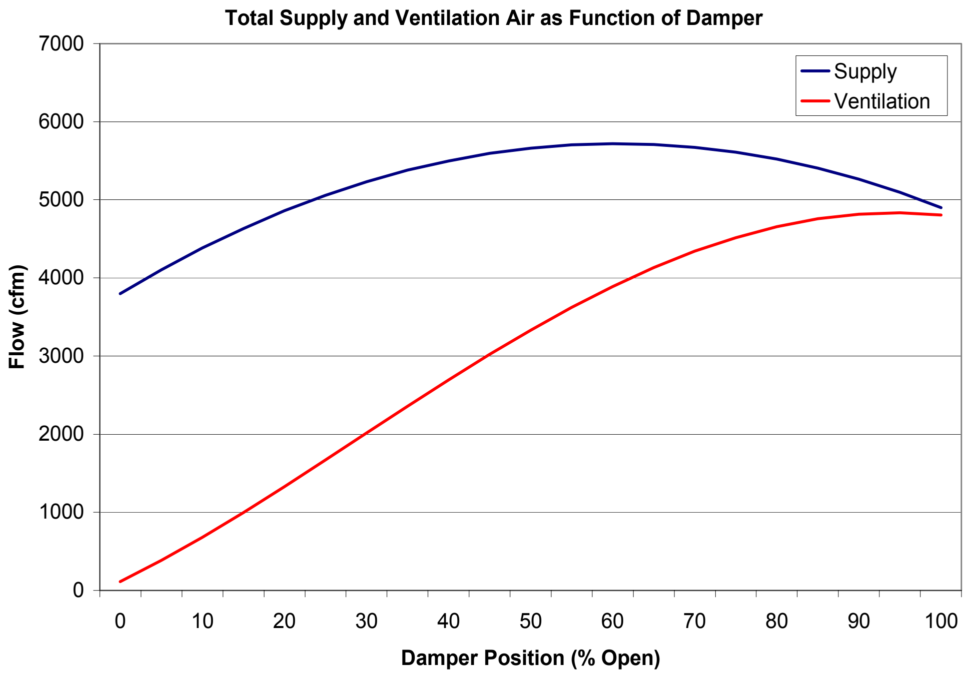


Figure 9. Supply and Ventilation Air Flow versus Damper Position for Watt Avenue (Sacramento) McDonalds Site

Calibrated VSAT Simulations

The major limitation of the blackbox modeling approach is that it not useful for extrapolating to drastically different weather conditions and control strategies (e.g., DCV On and Off) beyond those used during training. An alternative approach involves using VSAT to predict energy usage for the different strategies. Site specific information on the buildings and HVAC equipment were used as inputs to VSAT simulations. The VSAT model was calibrated using the same training data sets for the blackbox inverse models. The calibration process involved modification of parameters that are only known approximately. Examples of the parameters used to tune the VSAT model include:

- Window shading coefficient
- Window thermal resistance
- Window solar transmissivity
- Outdoor air infiltration rate

The calibrated VSAT models were then used along with 2002 weather data and the estimated occupancy based on CO_2 levels to compare cooling season performance for DCV On and Off. A description of how the occupancy, measured in terms of a net CO_2 generation rate, was determined is given in Appendix A. Simulation of the Milpitas site with the VSAT model required additional tuning adjustments, since the PlayPlace area has no external south facing walls. During the mid-day time frame there is little solar heat gain for the Milpitas site. The default VSAT model assumes that the wall and window areas are the same for each orientation, and thus adjustments were required.

Table 5 shows the performance of the calibrated VSAT models for the Bradshaw and Milpitas sites. Figure 10 shows sample comparisons between hourly predictions and measurements of compressor energy usage for the Bradshaw site. Although the accuracy of the VSAT model is not as good as the blackbox models for the training data, it is more useful for extrapolating performance.

Table 5. VSAT Model Performance Compared to Actual Measured Data During Cooling Season Training Periods for McDonalds PlayPlace Areas

Performance Metric	Bradshaw Road	Milpitas
Total energy usage, actual (kW-hr)	2100	714
Total energy usage, predicted (kW-hr)	2104	606
Root mean square error (hourly kW-hr)	1.79	1.28
Average error in the daily total energy used (kW-hr per day)	+0.09	-5.4
Root mean square error in daily total energy used (kW-hr)	15.3	8.9

Model calibration period same dates as for blackbox inverse model training

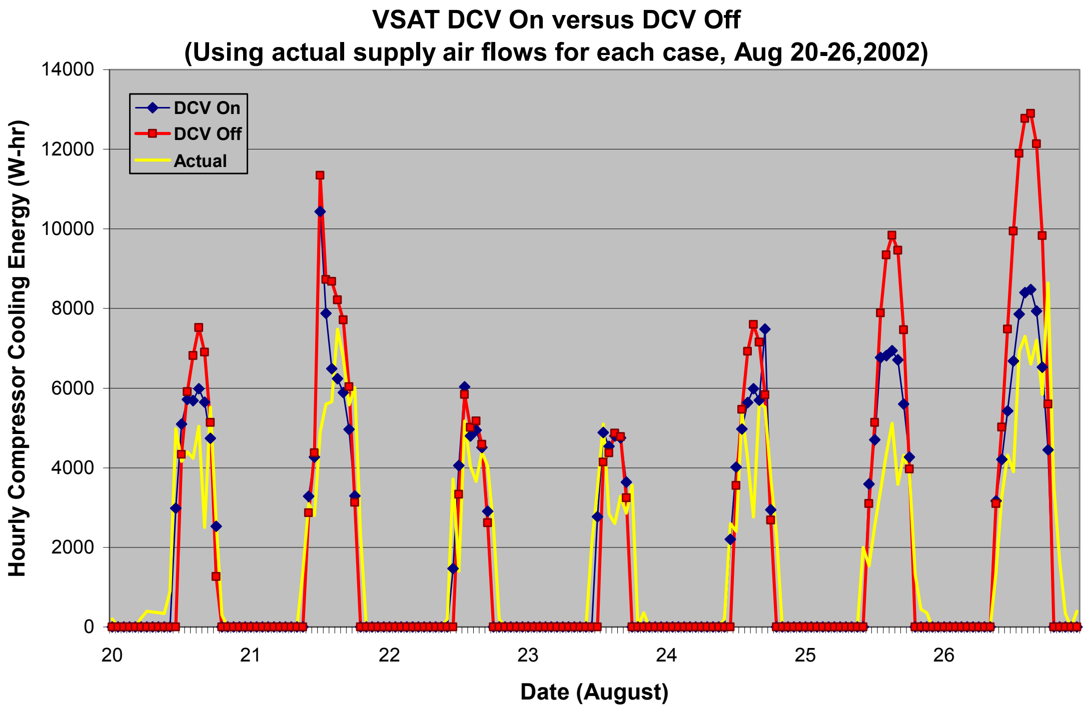


Figure 10. Hourly VSAT Model Predictions of Compressor Energy for DCV On and Off at Bradshaw (Sacramento) McDonalds Site for Period with DCV On

Table 6 gives comparisons of cooling season performance for DCV On and DCV Off at the Bradshaw and Milpitas McDonalds PlacePlaces obtained using the calibrated VSAT models. Similar results would be expected for the other Sacramento and Bay

Area PlayPlaces. The predicted savings, as well as the total condensing unit power demand, obtained with VSAT are lower than those determined using the blackbox inverse models. Part of the explanation may be that VSAT cannot model infiltration due to the opening of doors, which is highly variable. Infiltration tends to reduce CO_2 concentrations which would allow the DCV to provide less ventilation air and increases savings compared to fixed ventilation control.

Table 6. Cooling Season Energy Savings Predicted Using Calibrated VSAT Models for McDonalds PlayPlace Areas

	Bradshaw	Milpitas
Cooling season only fan energy (kWh)	7,604	8,150
Condensing unit, DCV On (kWh)	8,472	2,054
Condensing unit, DCV Off (kWh)	10,012	2,016
Total savings, condensing unit + fan energy	1,540	-38
% annual electrical energy savings, condensing unit alone	15.4%	-1.9%
% annual electrical energy savings, condensing unit + fan	8.7%	-0.3%
Savings in maximum hourly peak demand (peak DCV Off – peak DCV On, kW)	5.3	2.0

2.3 Modular Schoolrooms

Side-by-side comparisons for the modular school did not yield meaningful differences between DCV On and Off. Therefore, modeling approaches were used to compare these strategies for the same sites.

The Oakland schoolrooms have manually activated timers on all the HVAC units. The teacher activates the HVAC system (including supply fan) by turning a timer. The timer is only for one hour operation, so the teacher must reactive the timer each hour for continuous operation. It was observed that the units have only operated a few hours each school day. It appears that the amount of time the HVAC systems have operated each day has been nearly the same for both DCV On and DCV Off at both rooms at the

Oakland site. However, it's difficult to quantify operational differences and therefore side-by-side testing and hourly inverse modeling were not applied to this site. Calibrated VSAT simulations were used to estimate savings at the Oakland schools.

At the Woodland Gibson site, the rooms are set on programmable thermostats that turn the HVAC units on approximately 1 hour before full occupancy and off precisely as school is out at 3 pm. Inverse models were applied to this site.

Correlated Daily Energy Usage

Figures 11 and 12 show daily energy usage for cooling as a function of daily average ambient temperature for the the Woodland site (Sacramento area) for both DCV On and Off. The average daily cooling energy use is a nearly linear function of ambient temperature. However, there appears to be no real difference in energy usage regardless of the control strategy chosen for the Woodland site. The average damper position for DCV On is essentially the same for both strategies implying that the rooms are fully occupied most of the time when the HVAC system is on and design ventilation air is required to maintain the CO_2 setpoint for DCV On.

For the Woodland Gibson room 1, a special test was performed with the outdoor air damper set to match the amount of ventilation air provided with a unit that has a standard factory issue fixed louver configuration. The amount of ventilation air for this configuration is too small for the occupancy and is approximately 110 cfm or around 3 to 4 cfm per person. Therefore, typical installations for modular schoolrooms probably do not provide adequate indoor air quality (CO_2 concentrations are given in section 4). At this lower ventilation air flow rate, the energy usage was nearly the same for both DCV On or DCV Off.

Figures 13 and 14 give similar results for the Oakland schoolrooms. The data do not correlate nearly as well with daily ambient temperature as for the other sites. Although it appears that DCV On results in some energy savings, the differences are within the uncertainty of the correlation with ambient temperature.

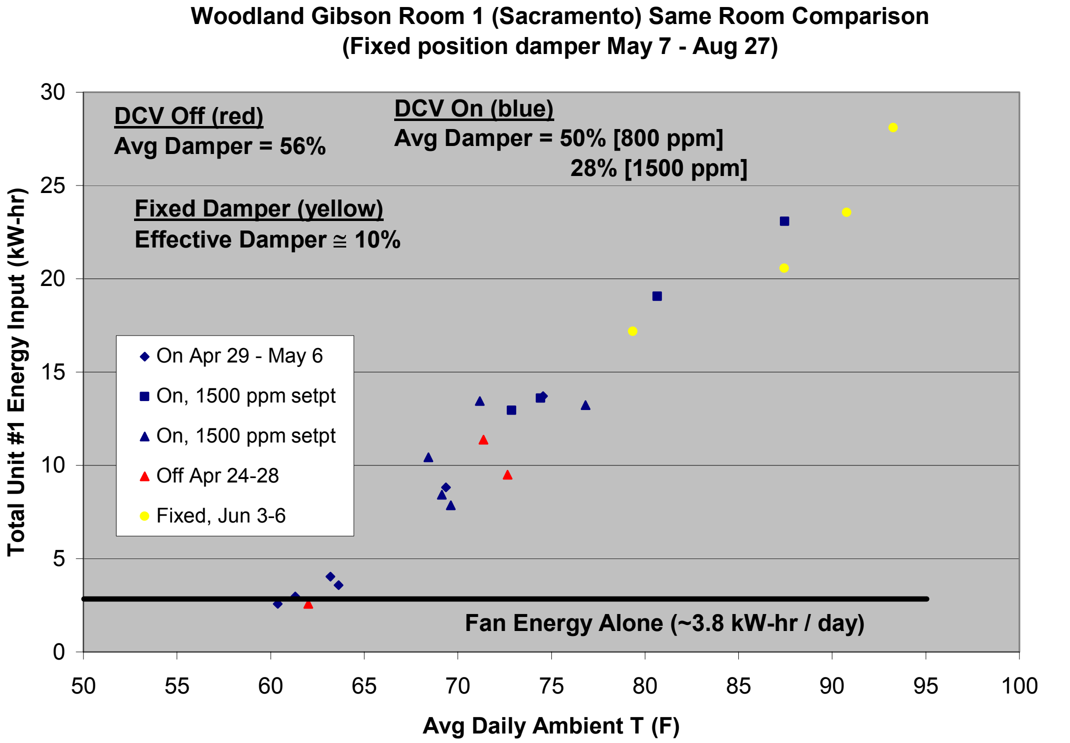


Figure 11. Correlated Daily Cooling Energy Use for DCV On and Off at Woodland (Sacramento) Schoolroom 1

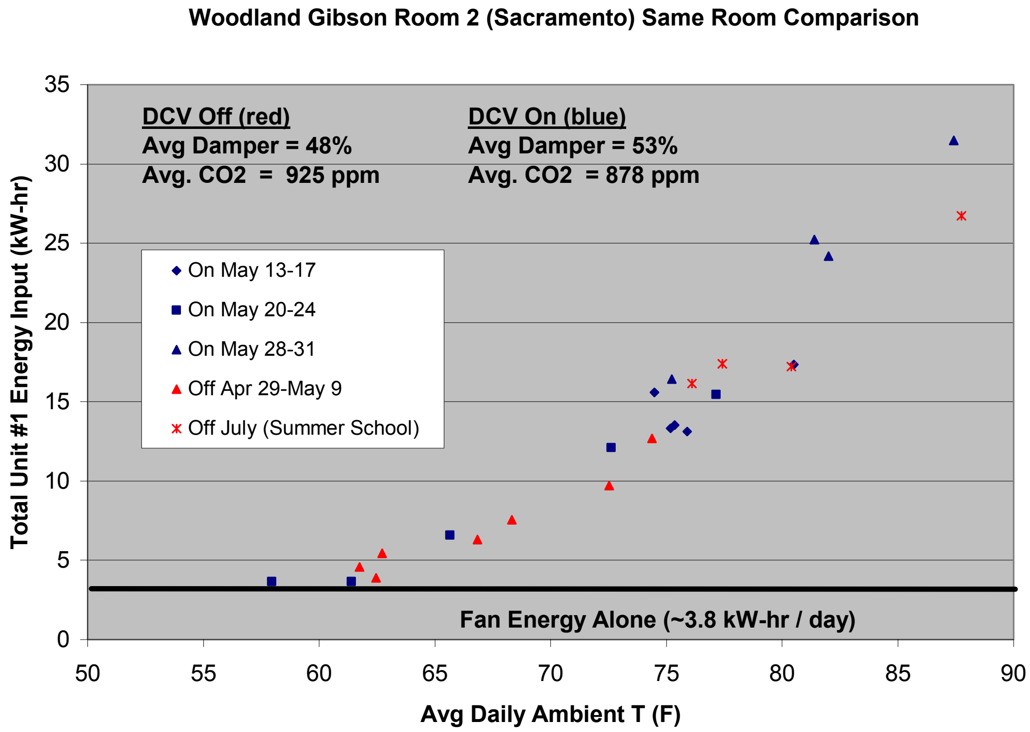


Figure 12. Correlated Daily Cooling Energy Use for DCV On and Off at Woodland (Sacramento) Schoolroom 2

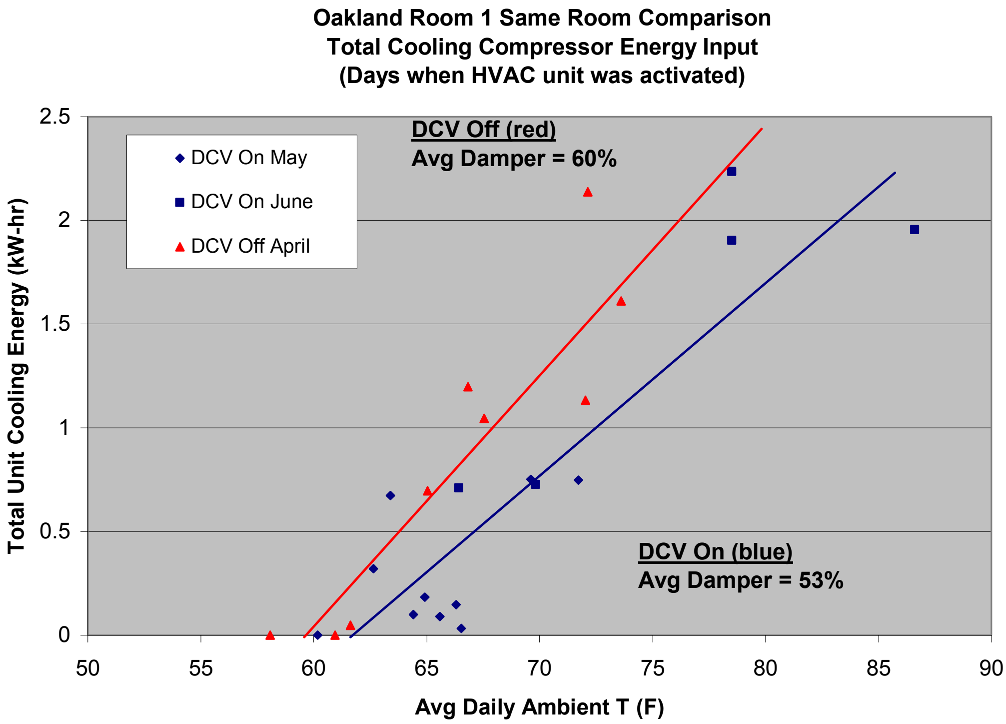


Figure 13. Correlated Daily Cooling Energy Use for DCV On and Off at Oakland Schoolroom 1

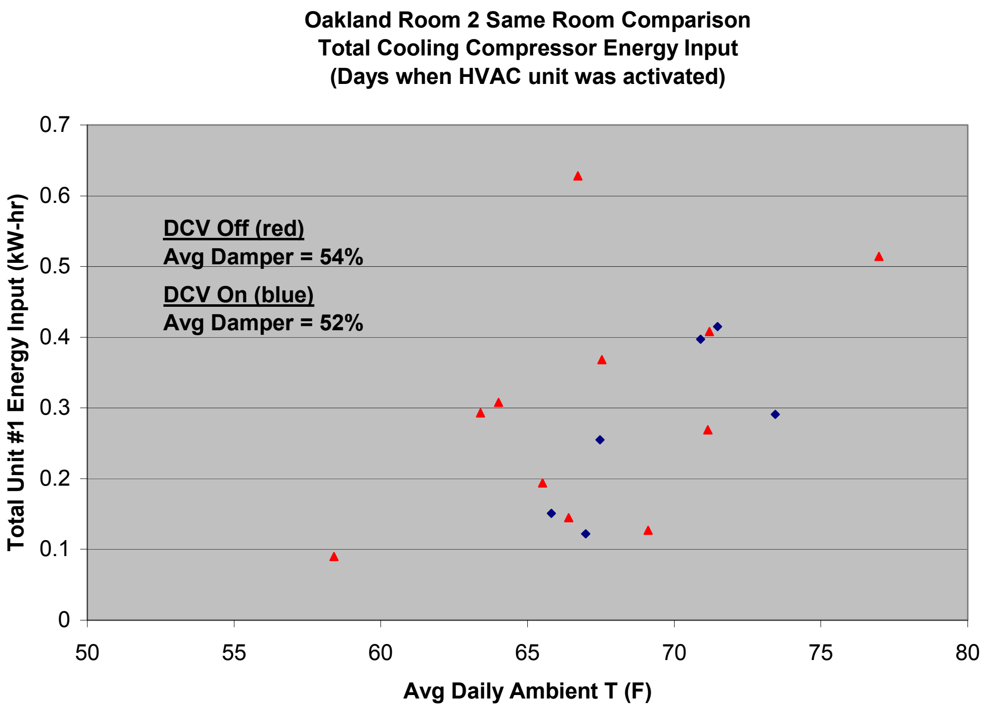


Figure 14. Correlated Daily Cooling Energy Use for DCV On and Off at Oakland Schoolroom 2

Correlated Hourly Energy Usage

The same inverse modeling approach applied to the McDonalds sites was used for the Woodland Gibson schoolrooms. During the majority of the cooling season data gathering period, the first Gibson schoolroom was set to have ventilation air flow to match the "factory standard" condition with fixed air louvers. This was part of an indoor air quality check and is described in more detail in Section 4 of this report. Thus, only limited data were available to develop models for DCV On and DCV Off with this room. During most of the cooling season period, the second room was set to DCV On for the IAQ comparison. More data were available to build comparison models for the Gibson room #2, and are presented here.

Table 7 summarizes the performance of the models for Gibson room 2 over the training period. The models did well in predicting the energy use. However, the model for DCV Off underpredicted the peak cooling demand. There was only one week of training data for the DCV Off operating mode with this room.

Table 7. Blackbox Model Performance Compared to Actual Measured Data During Cooling Season Training for Woodland Gibson Site

Performance Metric	Control Strategy	
Total kW-hr Energy Usage	DCV On DCV Off	-1.8% -2.8%
Root mean square error (hourly kW-hr)	DCV On DCV Off	0.46 0.51
Peak cooling energy demand	DCV On DCV Off	+1.2% -26.7%

Figure 15 gives hourly comparisons of measured compressor power for DCV On and predicted compressor power for both DCV On and Off during the week of May 13-17, 2002 in Woodland. Energy use for DCV On is slightly higher than for the DCV Off case. These results are consistant with those determined using the models for daily energy usage. Table 8 shows spring cooling season comparisons obtained with the blackbox models using weather data from 2002 and estimated occupancy patterns. The use of DCV for this site resulted in greater energy usage. However, as shown in section 4, the CO_2 levels were lower for DCV On than for DCV Off, implying that better indoor air quality was realized.

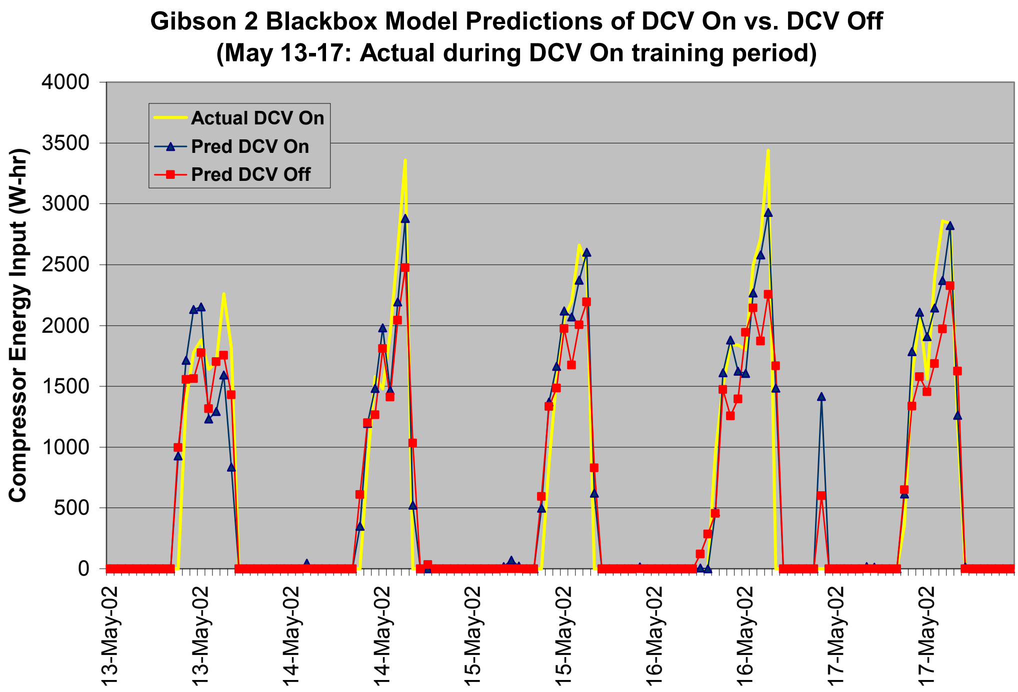


Figure 15. Hourly Blackbox Model Predictions of Compressor Energy for DCV On and Off at Woodland (Sacramento) Gibson Schoolroom 2 for Period with DCV On

Table 8. Spring Cooling Season Energy Savings Predicted Using Blackbox Models for Woodland Gibson Modular Schoolroom

	Gibson #2
Fan energy, DCV On (kW-hr)	176
Fan energy, DCV Off (kW-hr)	176
Compressor energy, DCV On (kW-hr)	903
Compressor energy, DCV Off (kW-hr)	801
Total savings, compressor energy only	-102
Total savings, compressor + fan energy	-102
% annual electrical energy savings, compressor + fan	-12.7%
% annual electrical energy savings, compressor + fan	-10.4%
Savings in maximum hourly peak demand (peak DCV Off – peak DCV On, kW-hr)	-0.6

Note: Spring cooling season was taken from late April through first week of June.

Calibrated VSAT Simulations

A VSAT model of the Woodland Gibson modular school rooms was calibrated using measured data. Table 9 summarizes the performance of the model for the training period. Figure 16 shows hourly comparisons of actual and predicted compressor energy use over a week. The model works reasonably well in predicting compressor usage, although the root mean square error (0.76 kW-hr for DCV On) is higher than with the blackbox model at 0.46 kW-hr. Figure 16 also includes a comparison of DCV On and DCV Off predictions. Consistant with the daily and blackbox correlations, Figure 16 indicates that there are relatively small differences between the two control strategies for this site.

Table 9. VSAT Calibration Cooling Model Performance Compared to Actual Measured Data During Training Periods for Woodland Gibson School

Performance Metric	Gibson #2
Total energy usage, actual (kW-hr)	713
Total energy usage, predicted (kW-hr)	712
Root mean square error (hourly kW-hr)	0.76
Average error in the daily total energy used (kW-hr per day)	-0.03
Root mean square error in daily total energy used (kW-hr)	4.8

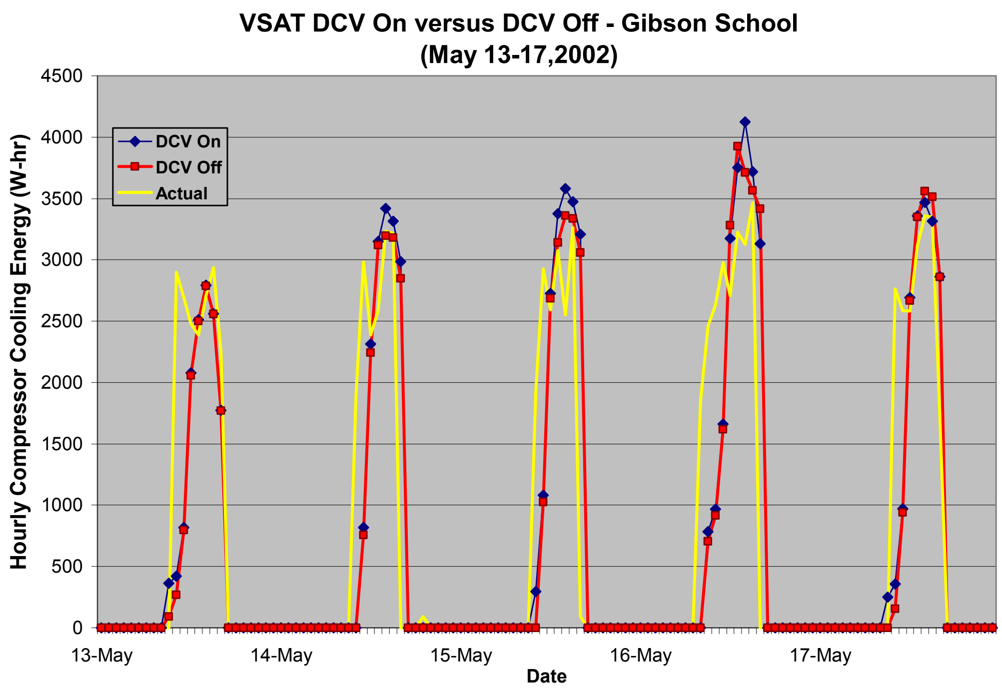


Figure 16. Hourly Calibrated VSAT Model Predictions of Compressor Energy for DCV On and Off at Woodland (Sacramento) Gibson Schoolroom 2 for Period with DCV On

Since all analytical comparion methods (correlated daily average, correlated hourly and calibrated VSAT models) indicate no real savings potential for the modular school rooms, the investigation also focused on comparisons to a system that has the factory issue fixed ventilation louver configuration. The standard factory issue Bard unit has outdoor air ventilation brought in through fixed louvers on the outer panel. Additional ventilation air can be provided if the optional controllable damper package is purchased. From discussions with school system personnel during the course of this study, it was estimated that less than half of the units are supplied with the controllable damper assembly. Without the controllable damper assembly, there is no possibility to take advantage of “free” cooling through the use of economizer control. This is particularly important for climate zones in California and for school applications that may have most of their actual operating time during the spring and fall cooling seaons.

Appendix B describes the testing done to measure the amount of ventilation air provided with the fixed louver configuration. This information was used to also simulate the system performance in VSAT. Table 10 shows total yearly results for the two control strategies along with results for the original installation having no economizer with fixed ventilation louvers. The annual energy usage for cooling is nearly identical for DCV On and DCV Off. However, the installation of the economizer resulted in about a 12% reduction in energy usage. Also, as shown in section 4, the addition of demand-controlled ventilation resulted in improved indoor air quality.

A VSAT model was also calibrated for the Oakland school. Since the Oakland schoolrooms have manual timers installed, the comparison with field data is not as complete as with the Gibson school. There were 14 schools days with the cooling on multiple hours in a row that were used to develop a calibrated VSAT model. During the 14 day training period, a total of 114 kW-hr of energy was measured for the condensing unit usage. The calibrated VSAT model gave a total of 107 kW-hr during this period and had a root mean square error of the hourly predicted values of 0.2 kW-hr.

Table 11 gives cooling season savings estimated with the VSAT model. The model estimates small positive savings for this site. In comparison with the Gibson school, the Oakland school has more variability in occupancy that results in greater savings potential. The Oakland school is a large city high school, while the Gibson site is a smaller city elementary school. It is also noted that the total amount of compressor cooling energy for the two climate type locations is nearly the same. During the normal school year, the two sites have fairly similar weather patterns and school is not regularly in session during the summer.

Table 10. Full Year Cooling Energy Savings Predicted Using Calibrated VSAT Model for Woodland Gibson School

	DCV plus economizer	Economizer alone	Basic factory issue system, no economizer or DCV
Fan energy, (kW-hr)	525	525	525
Condensing unit energy, (kW-hr)	2,750	2,748	3,137
Total savings, condensing unit energy only compared to basic factory issue HVAC system (kW-hr)	387	389	---
Total savings, compressor + fan energy, compared to basic factory issue HVAC system (kW-hr)	387	389	---
% annual cooling energy savings, condensing unit + fan compared to basic factory issue HVAC system	10.6%	10.6%	---
Savings in maximum hourly peak demand compared to basic factory issue HVAC system (kW-hr)	-2.1 (higher)	-1.1 (higher)	---

Table 11. Full Year Cooling Season Energy Savings Predicted Using Calibrated VSAT for Oakland School

	Oakland
Fan energy, DCV On (kW-hr)	523
Fan energy, DCV Off (kW-hr)	523
Compressor energy, DCV On (kW-hr)	2,594
Compressor energy, DCV Off (kW-hr)	2,711
Total savings, compressor energy only	117
Total savings, compressor + fan energy	117
% annual electrical energy savings, compressor + fan	4.3%
Savings in maximum hourly peak demand (peak DCV Off – peak DCV On, kW-hr)	-0.3 (higher)

3. Energy Savings for Heating

It was difficult to evaluate energy savings for the heating season because the heating loads for these sites are relatively small and thus the “signal to noise” ratio associated with the data analysis procedures is is relatively high. For instance, as a percentage of total heating load, the full occupancy heat gains due to people are approximately 25% for the modular schools (3.8 out of 14.8 kW), 35% at the Milpitas McDonalds (6.9 out of 19.6 kW) and 25% at the Bradshaw McDonalds (9.4 out of 34.7 kW). Thus, variations in the occupancy patterns or uncertainty in the occupancy measurements significantly affect the accuracy of the comparative analysis. Furthermore, errors associated with estimating the natural gas usage using digital control signals may be significant when compared to the magnitude of the heat loads.

3.1 McDonalds PlayPlace Areas

Correlated Daily Energy Usage

Figure 17 shows daily gas heater on time as a function of average daily temperature for the two control strategies at the Bradshaw McDonalds site in the Sacramento area. A similar comparison plot is given in Figure 18 for the Castro Valley site in the Bay Area. There is almost no difference in energy usage between the two strategies for either climate type. This is in sharp contrast to the cooling season results for this site, where the savings for DCV were significant for the Bradshaw site and measureable for the Castro Valley site.

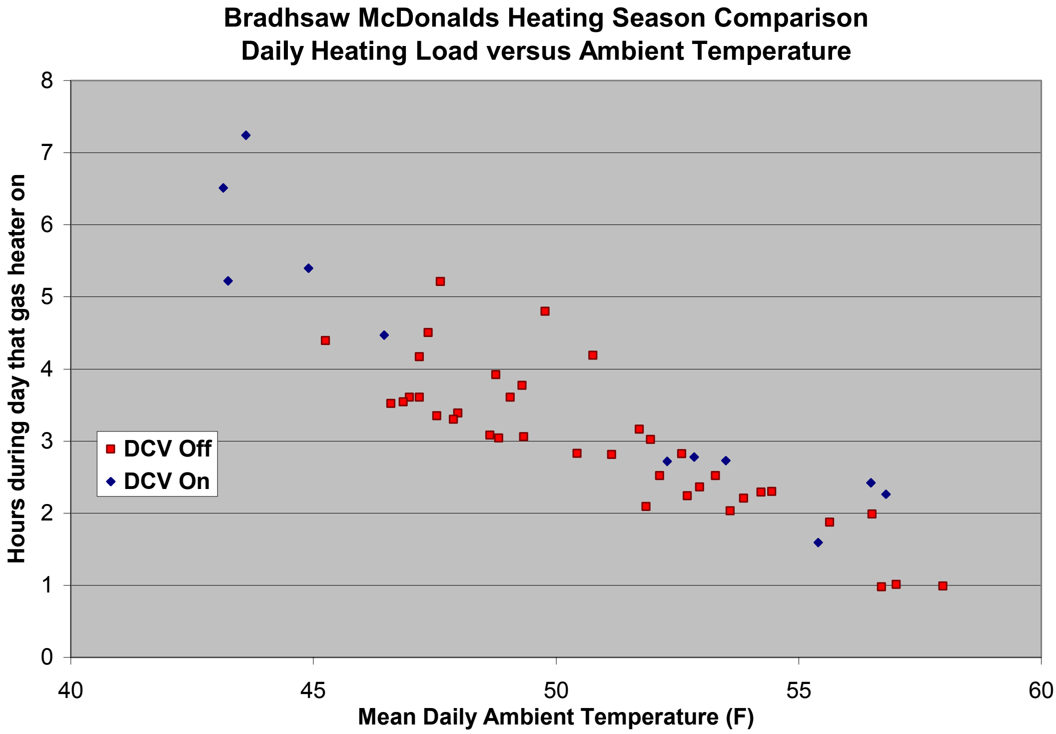


Figure 17. Daily Heater On Time as a Function of Average Daily Temperature for Bradshaw (Sacramento) McDonalds PlayPlace

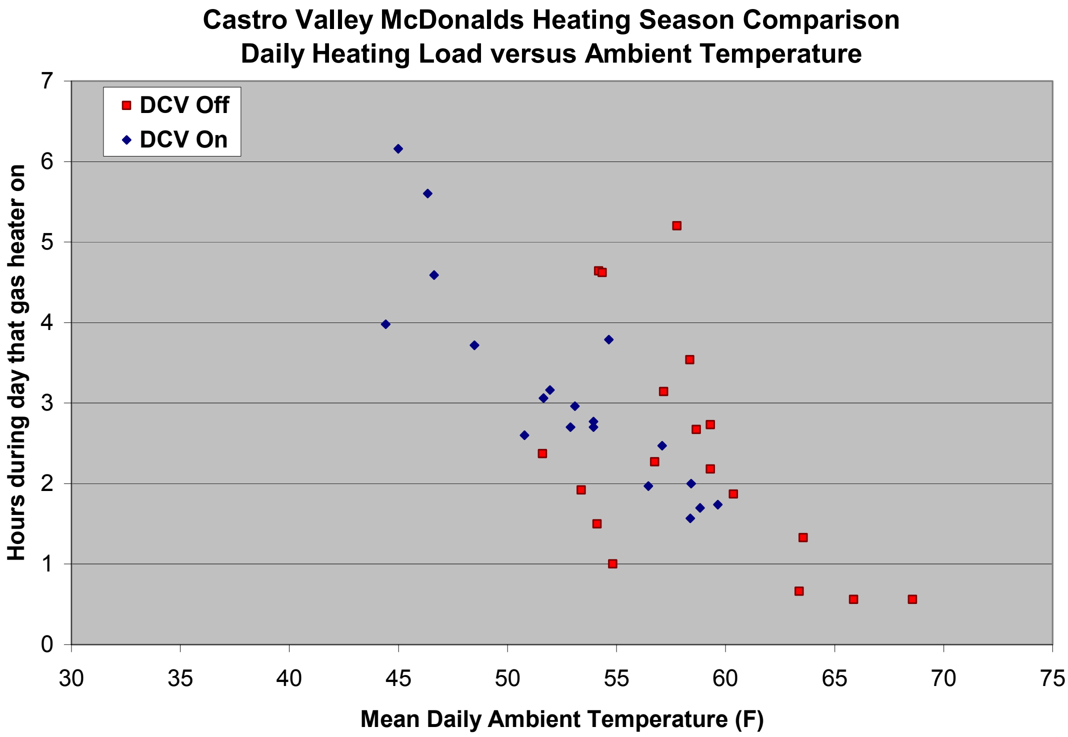


Figure 18. Daily Heater On Time as a Function of Average Daily Temperature for Castro Valley (Bay Area) McDonalds PlayPlace

Correlated Hourly Energy Usage

Blackbox models were developed for predicting hourly gas usage at the Bradshaw McDonalds site for both DCV On and Off. A total of about 25 days of data was available for training the models. Table 12 gives a summary of the performance of the models for the training data. Figures 19 and 20 give sample hourly comparisons of measured and predicted gas usage for two different weeks (one with DCV On and one with DCV Off). In both cases, the models do a good job of predicting hourly gas usage. Figures 19 and 20 also show comparisons between predicted gas usage for DCV On and Off over the two weeks. Consistant with the daily correlations, the differences in gas usage between the two strategies were relatively small for this site.

Since limited training data were available for the heating season, extrapolation of the heating results for the correlated hourly energy use models is not appropriate. Full season heating comparisons were done using the calibrated VSAT models in the following section of this report.

Table 12. Blackbox Model Performance Compared to Actual Measured Data During Heating Season Training Periods for McDonalds PlayPlace Areas

Performance Metric	Control Strategy	Bradshaw Road
Total Energy Usage (cu. ft. gas)	DCV On DCV Off	-1.0% -2.9%
Root mean square error (hourly cu. ft. gas usage)	DCV On DCV Off	7.6 5.5

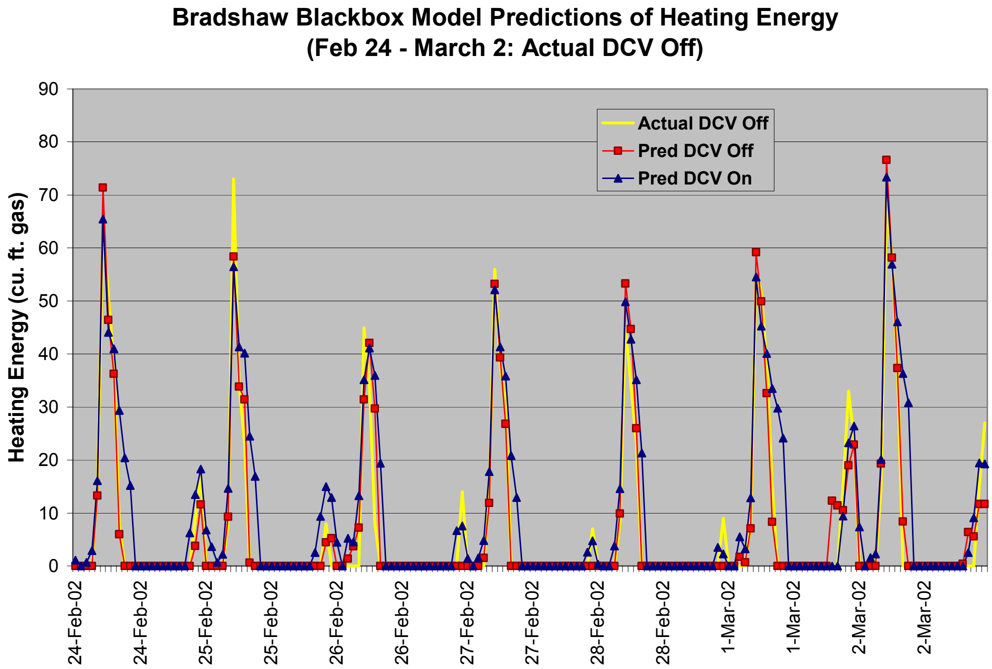


Figure 19. Hourly Blackbox Model Predictions of Gas Usage for DCV On and Off at Bradshaw (Sacramento) McDonalds PlayPlace for Period with DCV Off

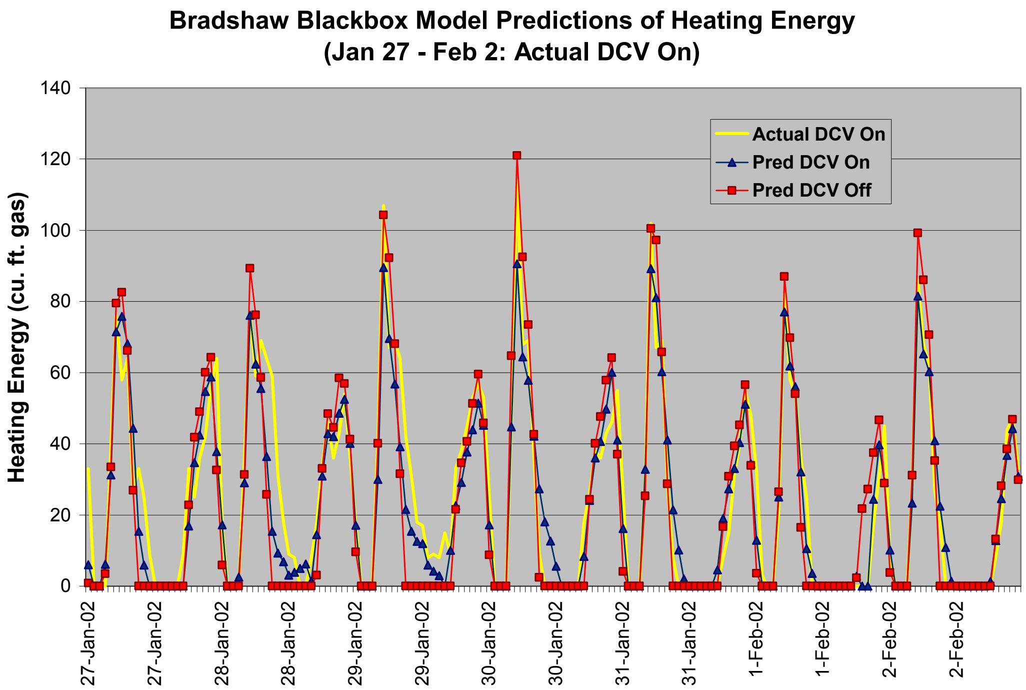


Figure 20. Hourly Blackbox Model Predictions of Gas Usage for DCV On and Off at Bradshaw (Sacramento) McDonalds PlayPlace for Period with DCV On

Calibrated VSAT Simulations

The calibrated VSAT model used for the cooling season analysis also provides estimates of energy usage during the heating season. Table 13 compares the energy saving predictions using the calibrated VSAT models for the Bradshaw (Sacramento) and Milpitas (Bay Area) McDonalds sites.

Table 13. Heating Season Energy Savings Predicted Using Calibrated VSAT Models for McDonalds PlayPlace Areas

	Bradshaw (Sacramento)	Milpitas (Bay Area)
Gas usage for heating, DCV On (therms)	518	106
Gas usage for heating, DCV Off (therms)	2,231	2,554
Total savings (therms)	1,713	2,448
% annual gas usage energy savings	77%	96%

The percentage savings are larger than for cooling, because the ventilation load is a larger fraction of the total load. This is generally the case when comparing cooling and heating results. While the percentage savings predicted by the calibrated VSAT routines are impressive, the absolute magnitudes of the savings are small. The total costs of heating for these systems are relatively small compared to the cooling costs. The fact that the field data did not show savings can be attributed to a small "signal to noise ratio" in the heating data. The magnitude of the heating energy usage is relatively small and significantly influenced by variations in occupancy, infiltration from door openings, etc. At small heating loads, these "random" events have as much impact as whether the system is operating in DCV On or Off mode.

3.2 Modular Schoolrooms

Correlated Daily Energy Usage

Figure 21 shows daily electrical energy usage for heating as a function of daily average ambient temperature for DCV On and Off at the modular schoolrooms at the

Woodland Gibson site. The systems utilize a heat pump for heating. The data do not correlate very well with ambient temperature. Although there appears to be savings associated with the use of DCV for this site, the uncertainty in the correlations is of a similar magnitude as the differences.

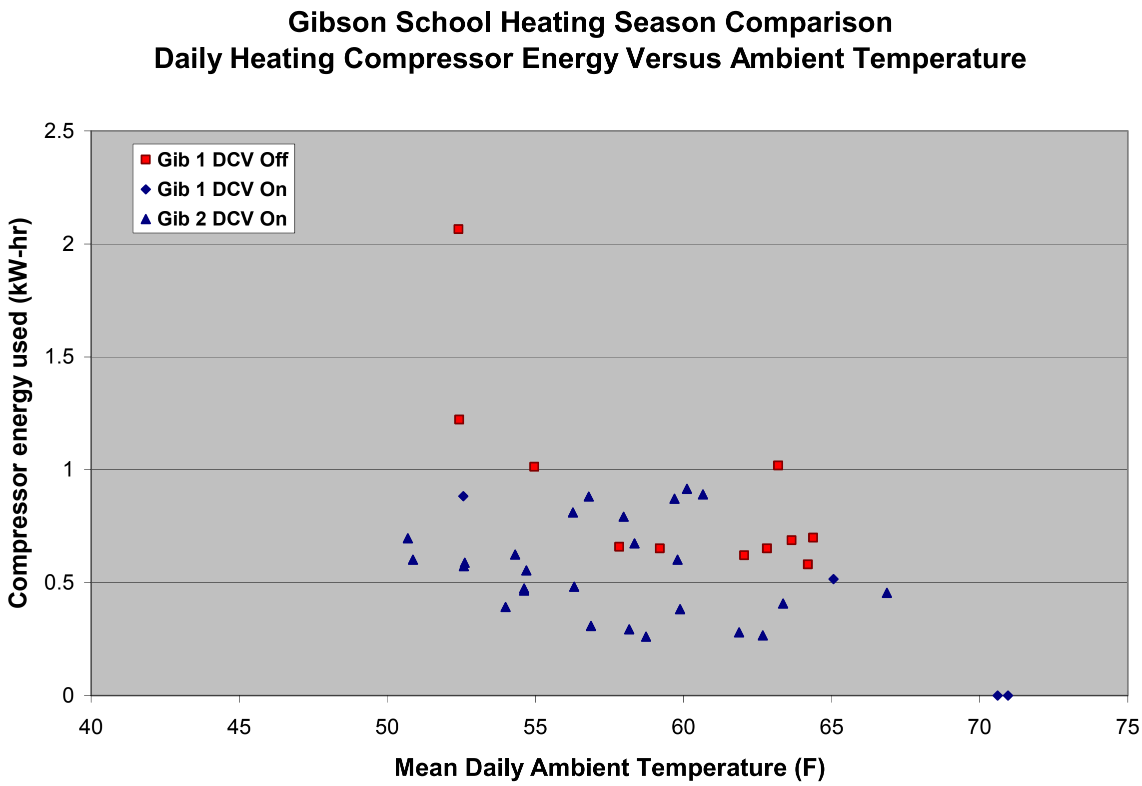


Figure 21. Daily Compressor Energy Usage for Heating as a Function of Average Daily Temperature for Woodland (Sacramento) Gibson Modular Schoolrooms

Calibrated VSAT Simulations

Just like with the McDonalds sites, the calibrated VSAT model used for the cooling season analysis also provides estimates of energy usage during the heating season. Table 14 compares the energy saving predictions using the calibrated VSAT models for the Gibson (Sacramento area) and Oakland school sites.

Table 14. Heating Season Energy Savings Predicted Using Calibrated VSAT Models for Modular Schoolrooms

	Gibson (Sacramento)	Oakland (Bay Area)
Total unit energy consumption, DCV On (kW-hr)	1922	1827
Total unit energy consumption, DCV Off (kW-hr)	2340	2030
Total savings (kW-hr)	418	203
% annual electrical energy savings, condensing unit + fan	17.9%	10.0%

4. Comparisons of Indoor CO_2 Concentrations

Controllers at the field sites are configured to maintain a setpoint for CO_2 concentration in the return duct with a specified minimum damper position. For the DCV On results, the setpoints were 800 ppm and the minimum damper positions were fully closed. For DCV Off, setpoints of 1500 ppm were established and the minimum damper positions were set to values that provide ventilation air matching the requirements of ASHRAE 62-1999. This section presents comparisons of return duct CO_2 for the two control strategies at several sites.

4.1 McDonalds PlayPlaces

Table 15 shows comparisons of average return air CO_2 concentrations during occupied periods for DCV On and DCV Off during the 2002 cooling season. The use of DCV results in higher CO_2 concentration levels for these test sites due to lower ventilation rates. This is consistant with the energy savings for DCV at these sites. The largest differences in CO_2 concentrations occur at the Bradshaw. Recall that this site also had the largest energy savings for DCV. The Bradshaw site has lower average CO_2 concentrations for DCV Off than the other sites, implying that the occupancy is lower at this location. Lower occupancies relative to design occupancies generally lead to larger energy savings for DCV.

Table 15. Mean CO_2 Levels with DCV On and DCV Off Control Strategies at McDonalds PlayPlace Areas

DCV Control Strategy	Bradshaw Road (Sacramento)	Milpitas (Bay Area)	Castro Valley (Bay Area)
Off	496 ppm	541 ppm	572 ppm
On	575 ppm	613 ppm	615 ppm

Figures 22-24 are histograms of the occupied hours that CO_2 concentrations fell within different bands for the Milpitas, Bradshaw, and Castro Valley sites. At the Milpitas and Bradshaw sites, the DCV controller was generally able to keep the return air CO_2 concentration at or below the 800 ppm setpoint. However, at the Castro Valley site, about 5% of the occupied hours were at CO_2 levels above 900 ppm.

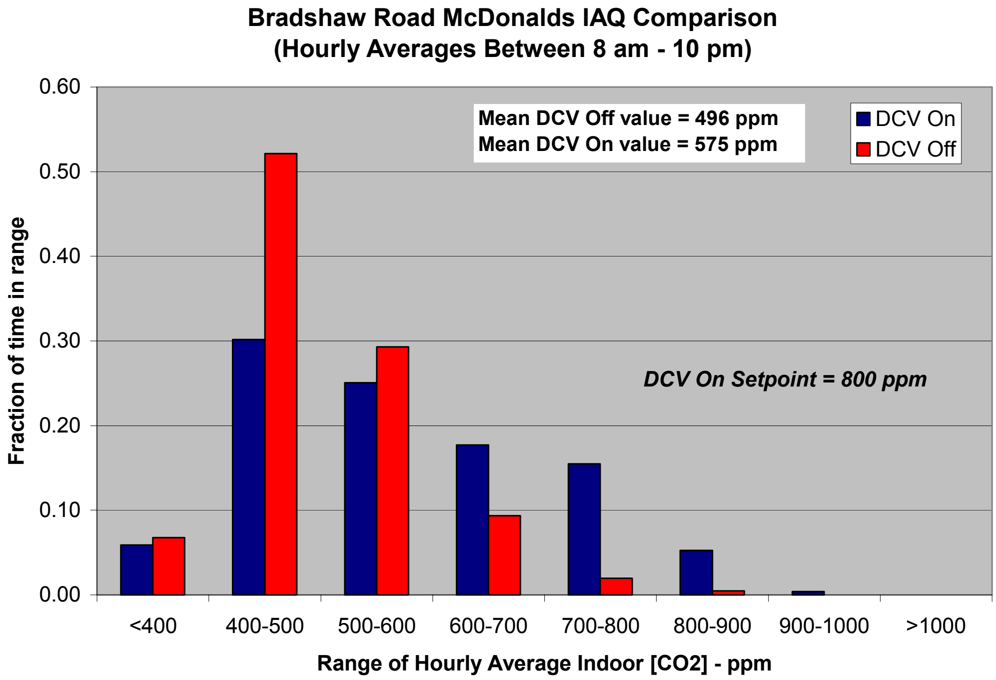


Figure 22. Histogram of Return Air CO_2 Concentrations at Bradshaw (Sacramento) McDonalds PlacePlace for DCV On and Off

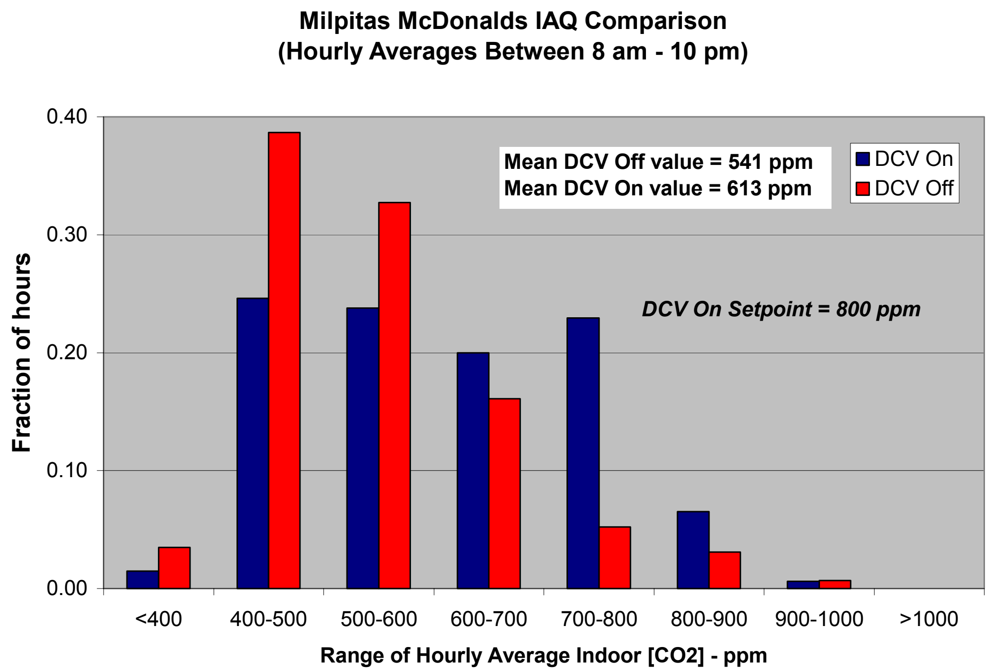


Figure 23. Histogram of Return Air CO_2 Concentrations at Milpitas (Bay Area) McDonalds PlacePlace for DCV On and Off

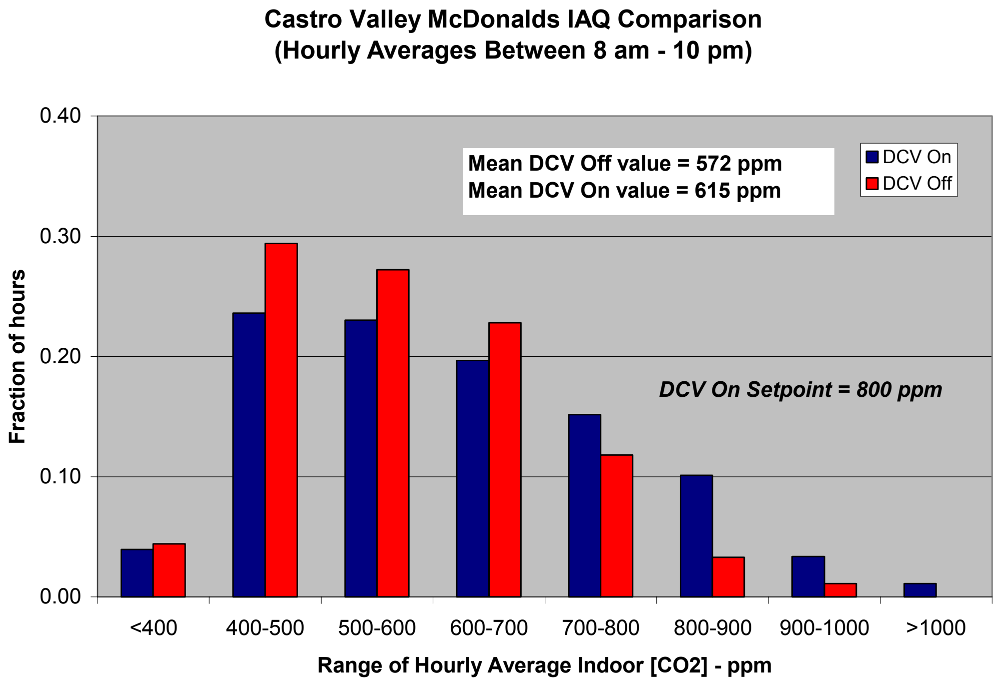


Figure 24. Histogram of Return Air CO_2 Concentrations at Castro Valley (Bay Area) McDonalds PlacePlace for DCV On and Off

4.2 Modular Schoolrooms

Figures 25 is a histogram for return air CO_2 levels at one of the Gibson schoolrooms. Results are included for DCV On, DCV Off with fixed ventilation satisfying ASHRAE Standard 62-1999, and DCV Off with the ventilation airflow at the same level measured at a similar room that has only fixed air inlet louvers. Fixed air inlet louvers are the standard factory configuration for the sidewall mounted HVAC units, unless the economizer option is purchased with a modulating outdoor air damper. Since this is an additional option to the HVAC package, it is probably not installed in most school rooms.

The results in Figure 25 imply that the use of DCV results in better indoor air quality than for fixed ventilation determined according to ASHRAE Standard 62-1999. Possibly the metabolic rates assumed for application of the standard are lower than actually occur for this application. Furthermore, the use of the “Factory Standard” installation results in very high CO_2 concentrations. Over 60% of the occupied hours with the Factory Standard configuration had CO_2 levels that exceeded 1200 ppm. These levels violate California Title 24 requirements. Appendix B describes the test procedure used to determine the airflow and CO_2 levels for a "factory standard" configuration and provides a more detailed discussion of the results for this particular study.

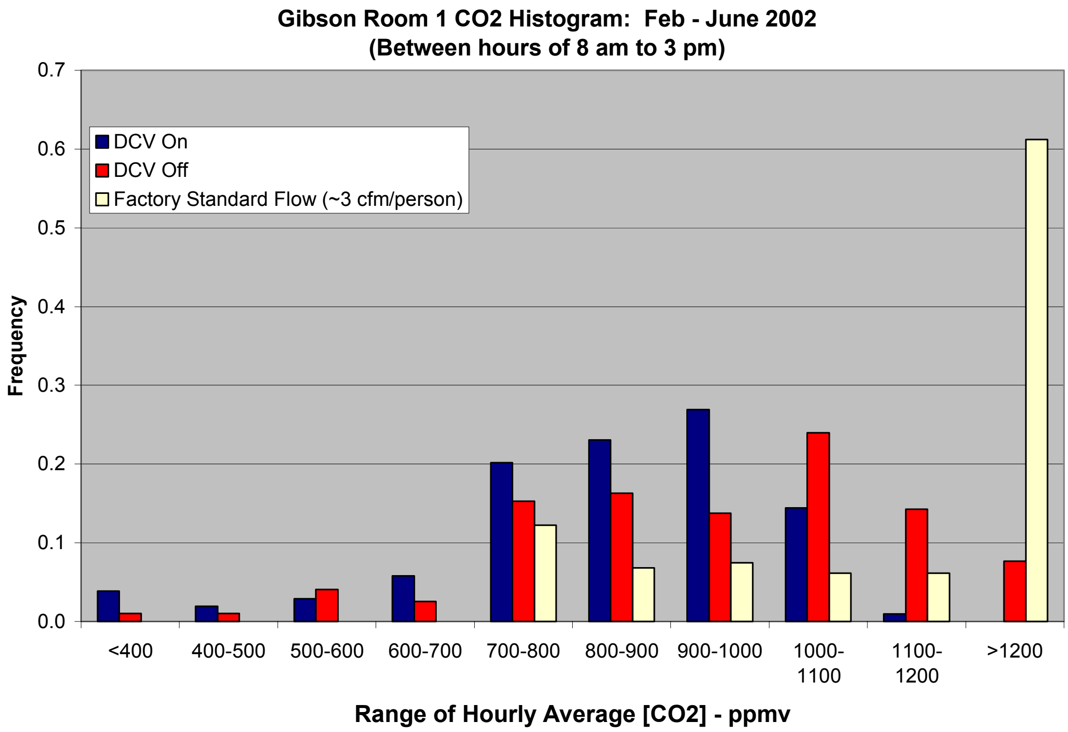


Figure 25. Histogram of Return Air CO_2 Concentrations at Woodland (Sacramento) Gibson Schoolroom 1 for DCV On, DCV Off, and Original Factory Installation

Figure 26 gives a histogram for the second Gibson schoolroom. Compared to room 1, the CO_2 levels are much higher for this room, implying a higher occupancy. However, there is a large number of hours for CO_2 concentrations above 1200 ppm with DCV Off that can't be explained by higher occupancy. This result may be due to problems with the controller. In some of the field sites, the minimum position for the outdoor air damper changes randomly at times and is not always maintained at the 40% setpoint for DCV Off.

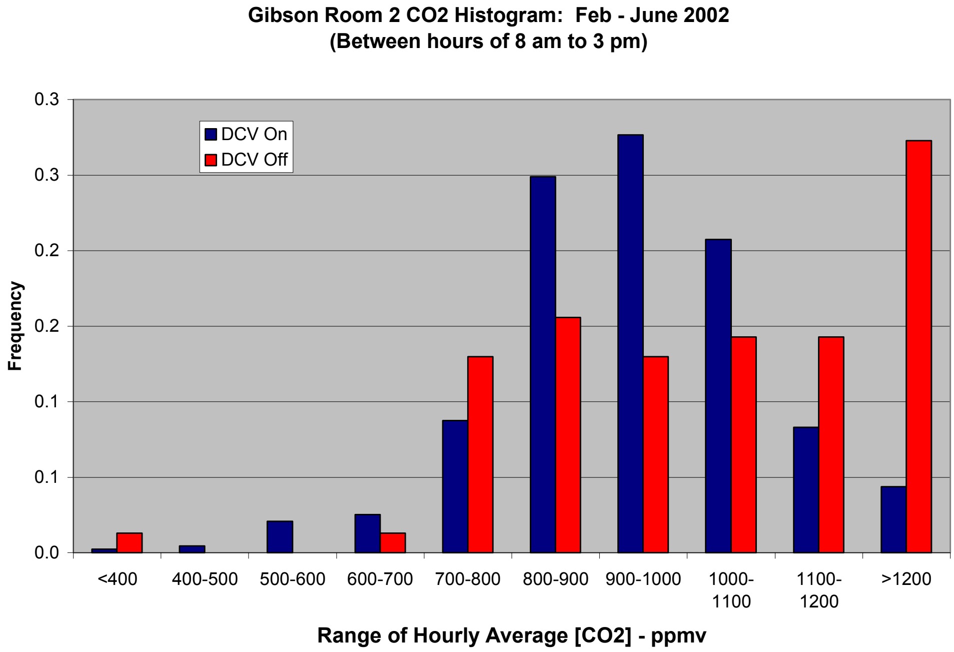


Figure 26. Histogram of Return Air CO_2 Concentrations at Woodland (Sacramento) Gibson Schoolroom 2 for DCV On and DCV Off

5. Summary and Conclusions

This report summarized initial cooling and heating season assessments of demand controlled ventilation (DCV) for California field test sites. The baseline of comparison was fixed ventilation rates based upon ASHRAE Standard 62-1999 (ASHRAE, 1999). Each of the sites has a differential enthalpy economizer and is capable of operating with and without DCV.

Both the energy usage and CO_2 concentrations were compared for the different control strategies. For energy analyses, several different approaches were developed for evaluating savings, including side-by-side comparisons, correlating daily energy usage, correlating hourly energy usage, and calibrating VSAT simulation models. In general, all the approaches gave consistant results. However, VSAT tended to underestimate savings for the PlayPlaces compared to the other more empirical approaches. Part of the explanation may be that VSAT cannot model infiltration due due to the opening of doors, which is highly variable. Infiltration tends to reduce CO_2 concentrations which would allow the DCV to provide less ventilation air and increases savings compared to fixed ventilation control.

For cooling, greater energy savings were achieved at the McDonalds PlayPlaces than for the modular schoolrooms. Primarily, this is because the PlayPlaces have more variability in their occupancy than the schoolrooms. The largest energy savings were achieved at the Bradshaw McDonalds PlayPlaces, which appears to have the lowest average occupancy level compared to the other McDonalds PlacePlaces. This site is located in Sacramento and has larger ventilation and total cooling loads than the bay area McDonalds. The savings in condensing unit energy were 35% and 16% for the Bradshaw and Milpitas sites, respectively. The total annual air conditioning cost savings were smaller (23% and 6%, respectively) because the supply fans operates continuously during occupied times for both strategies and fan energy is a significant fraction of the total energy usage.

There were no substantial cooling season savings for the modular school rooms, although the calibrated VSAT model for the Oakland school site does indicate some small (about 4%) savings The occupancy for the schools is relatively high with relatively small variability. The school sites are also on timers or controllable thermostats that mean the HVAC units only operate during the normal school day. The schools are also generally unoccupied during the heaviest load portion of the cooling season. Furthermore, the results imply that the average metabolic rate of the students may be higher than the value used in ASHRAE Standard 62-1999 to establish a fixed ventilation rate. In fact, the DCV control resulted in lower CO_2 concentrations than for fixed ventilation rate in the Woodland modular schoolrooms. It was not possible to confirm this trend at the Oakland schools because of the existence of manually activated timers on the HVAC equipment.

The amount of heating required for the California sites is relatively small and therefore absolute savings are relatively small for application of DCV. However, very large relative savings were estimated using calibrated VSAT predictions. These savings were not confirmed through direct comparison of field site data because the heating loads are small and the impact of random variations (e.g., occupancy) is as large as the effect of the control strategy at these small loads (i.e., a small signal-to-noise ratio). Overall, the total costs for providing heating at these sites is smaller than for cooling, so percentage savings are more important for the cooling cases.

6. References

ASHRAE. 1999. ANSI/ASHRAE Standard 62-1999: *Ventilation for acceptable indoor air quality*. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, Georgia.

Schuldt, M.A. and J.S. Romberger. 1998. Alternative approaches to baseline estimation using calibrated simulations. *ASHRAE Transactions* 104(2):871-879.

U.S. Department of Energy. 1997. *International performance measurement and verification protocol.* DOE/EE-0157. Washington, D.C.

APPENDIX A

Determination of net CO_2 generation rates in the field test buildings

The method for estimating the CO_2 source generation is based on a simple first-order mass balance. Using a mass balance for CO_2 in the occupied zone, the rate of change in the amount of CO_2 in a room is given by

$$\left.\frac{d[CO_2]}{dt}\right|_{room} = CO_2\big|_{\text{generated}} - CO_2\big|_{\text{removed}} - \overset{\sim 0}{CO_2\big|_{\text{stored}}} \tag{A.1}$$

For a given finite time period, the bulk mean concentration of CO_2 in the room can then be computed using

$$\left.\frac{d[CO_2]}{dt}\right|_{room} \approx \left.\frac{\Delta CO_2}{\Delta t}\right|_{room} = \frac{[CO_2]_i - [CO_2]_{i-1}}{\Delta t} \tag{A.2}$$

In Equation A.2, the subscripts *i* and *i-1* refer to the values during the current and previous time steps. The amount of CO_2 removed (in liters) from the room during a time period is determined by multiplying the amount of fresh outdoor air provided to the room by the concentration difference between the return air stream and the ambient.

$$CO_2\big|_{removed} = CO_2\big|_{\text{removed, supply air flow}} + CO_2\big|_{\text{removed, infiltration}} \tag{A.3}$$

$$\begin{aligned} CO_2\big|_{\text{removed, supply air}} &= \text{O.A. flow}\left(\frac{\text{ft}^3}{\text{min}}\right) * \\ &\left(\frac{[CO_2]_{\text{return, i-1}} - [CO_2]_{\text{ambient,i-1}}}{1\text{x}10^6}\right) * 28.3\,\frac{\text{liters}}{\text{ft}^3} * \ \Delta t\ (\text{min}) \end{aligned} \tag{A.4}$$

The outdoor airflow rate is determined from a curve fit of the measured field data for airflow rate as a function of damper position.

The amount of CO_2 that leaves the zone through infiltration or exfiltration with the surroundings can be highly variable, and is dependent on such factors as door or window openings, local wind velocity and direction and temperature differential between indoor and the ambient. For this analysis, the CO_2 removed through infiltration will be left as an unknown parameter and lumped in with the generation rate.

Rearranging Equation 3.4 for the CO_2 generation term gives

$$CO_2\big|_{\text{generated}} = \frac{d\,[CO_2]}{dt}\bigg|_{room} + CO_2\big|_{\text{removed, supply air flow}} + CO_2\big|_{\text{removed, infiltration}} \tag{A.5}$$

Since the amount of CO_2 that leaves due to infiltration is left as an unknown parameter, it is lumped into the generation rate term to provide a determination of the net CO_2 source generation. Doing so allows for an indirect accounting for some of the contributing events, such as door openings, which could occur at some regular pattern.

$$\begin{aligned} CO_2\big|_{\text{generated, net}} &= \left(CO_2\big|_{\text{generated}} - CO_2\big|_{\text{removed, infiltration}}\right) \\ &= \frac{d\,[CO_2]}{dt}\bigg|_{room} + CO_2\big|_{\text{removed, supply air flow}} \\ &= \frac{\Delta\,[CO_2]}{\Delta t}\bigg|_{room} + CO_2\big|_{\text{removed, supply air flow}} \end{aligned} \tag{A.6}$$

The above model development is based on knowing the changes in the bulk room CO_2 concentration. Direct physical measurement of this value requires measurement of the concentration at a large number of locations throughout the room during their occupied period, which is not practical for the field sites.

The bulk room concentration is not necessary if the assumption is made that the rate of change in the concentration of CO_2 in the return air tracks well the rate of change in the bulk room concentration. This model approach ignores the time lag between the time that a given mass of CO_2 is generated by the occupants in a room and the time that the mass reaches the return air grill. This time lag is dependent on the room volume, supply air flow rate, and the room layout. Since the time lag is ignored, then changes in the bulk room and return air concentrations are directly related. The time rate of change in the unknown bulk room concentration can be therefore be estimated from the time rate of change in the known return air concentration.

$$\frac{\Delta CO_2}{\Delta t}\bigg|_{room} \approx \frac{\Delta CO_2}{\Delta t}\bigg|_{room} = \frac{[CO_2]_{return,i} - [CO_2]_{return,i-1}}{\Delta t} \tag{A.7}$$

The above equations and discussion assume that the room is in equilibrium with respect to the CO_2 mass balance.

The net CO_2 generation rate in liters per second is then found for a given site from the generalized equation shown as A.8.

$$\begin{aligned}CO_2\big|_{\text{generated, net}} &= \left(CO_2\big|_{\text{generated}} - CO_2\big|_{\text{removed, infiltration}} \right) \\ &= \text{Room Volume} \cdot \frac{[CO_2]_{return,i} - [CO_2]_{return,i-1}}{\Delta t} \\ &\quad + \text{Ventilation airlfow rate} \cdot \left([CO_2]_{return} - [CO_2]_{ambient} \right)\end{aligned} \tag{A.8}$$

APPENDIX B

Comparison of Internal CO_2 Concentrations at a Modular Schoolroom with Factory Installed Fixed Ventilation Amount to One with Modulating Dampers and Demand Controlled Ventilation

Background

As part of the ongoing Building Energy Efficiency Program projects sponsored by the California Energy Commission, demand controlled ventilation (DCV) systems were installed at two school districts in California. Each of the four rooms where DCV was installed did not originally have controllable outdoor air dampers, which would be required for any DCV or economizer package installation. This appears to be the rule rather than the exception for typical modular schoolroom installations, since a controllable damper is an additional cost option for the HVAC package units provided by Bard Manufacturing. Initial cost is a primary driver in the decision process for these facilities since many of the schoolrooms are leased from outside entities, or are purchased by school districts with limited capital budgets.

This report provides a brief summary of a recent study measuring internal CO_2 levels in a room with fresh ventilation airflow set equal to a typical room with the standard fixed ventilation louver. The airflow rate is set by fixing the outdoor air damper at a position to provide the same flow as the standard room. This airflow rate is less than the values specified by ASHRAE or California Title 24. The results are compared to a room with an operating DCV system.

Test Site Description

The HVAC service technician and energy management specialist at the Woodland School District agreed to help set up on of our test rooms to match the outdoor air ventilation flow of a room with the standard fixed ventilation louvers. Woodland is approximately 20 miles west of Sacramento. The test rooms are side-by-side installations of two rooms that stand somewhat isolated from the remainder of the modular schoolrooms at this site (Figure B-1). During the 2001-02 school year, sixth grade students occupied these rooms with 32 students per room.

Figure B-1 Woodland School Site Test Rooms – Rear View Looking East

Even with the modulating outdoor air dampers fully opened, the amount of fresh ventilation air is less than 60% of the total system flow, since the damper does not block the return air stream. Figure B-2 shows a plot of the regression equation derived from field test measurements made on the units at Woodland.

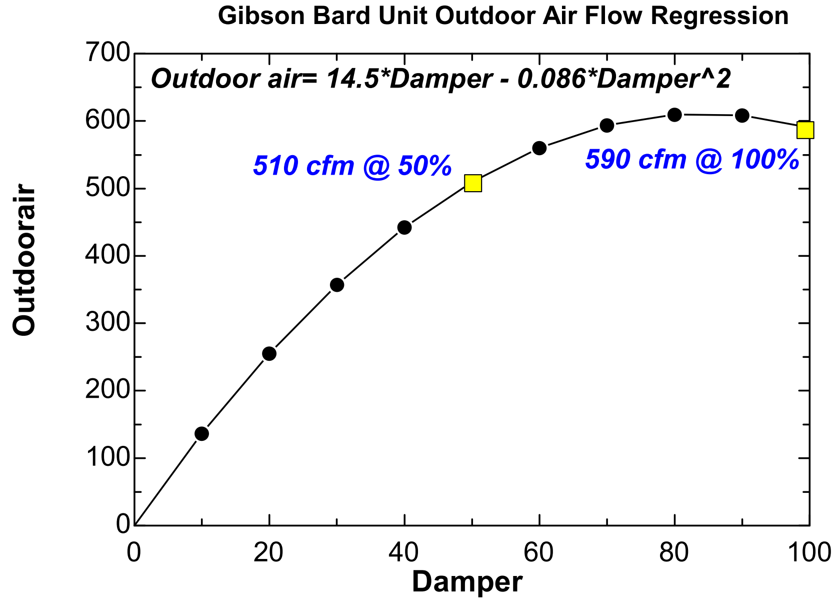


Figure B-2 - Outdoor Airflow for Room as Function of Damper Position

Ventilation Air Test Setup

The basic test plan approach was to first measure the amount of outdoor air ventilation flow at an identical room at the site with a heat pump unit that has the fixed louver configuration. (All other rooms at this site are with fixed louvers.) The outdoor air damper in the test room was then fixed in place such that the ventilation air matched that for the fixed louver configuration.

Ventilation airflow readings were made using an airflow hood provided by the HVAC tech at the school system.

Typical Modular Schoolroom with Fixed Ventilation Louver

The system airflow balance was checked at a nearby room with standard factory installed fixed ventilation louvers. The results are summarized in Figure B-3 below.

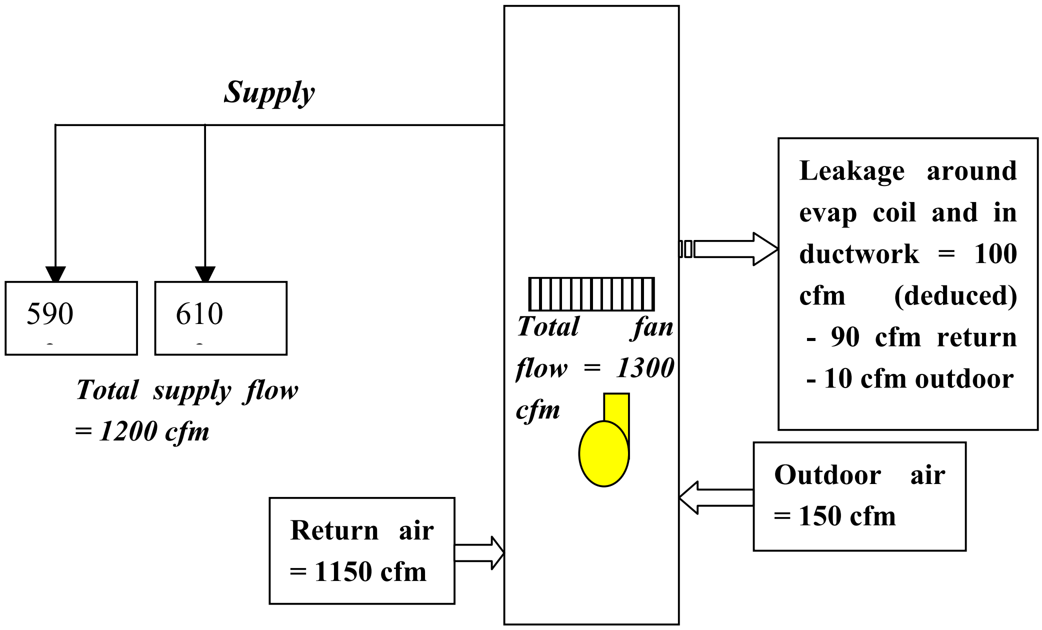


Figure B-3 - Airflow Balance for Typical Room Without Controllable Damper

In this configuration:

- Total supply airflow = 1200 cfm
- Total return airflow at grill = 1150 cfm
- Outdoor air flow into unit = 150 cfm
- Total supply fan flow (return + OA) = 1300 cfm
- Leakage in evaporator coil, ductwork (deduced) = 100 cfm
  (assume 10 cfm from outdoor air and 90 cfm from return)
- Source of supply airflow (assuming complete mixing in fan)

Return air	1060 cfm
Outdoor air =	140 cfm
TOTAL	1200

- % Outdoor air to room = $\frac{140}{1200} = 11.7\%$

The supply airflow measured for the schoolrooms in this test program ranged from 1000 to 1200 cfm. The manufacturer's rating for sidewall heat pump unit of this size is 1400 cfm. The total supply flow is a little lower due to the additional pressure drop of the supply ducting inside the room and/or duct leakage.

Test Site Modular Schoolroom Modification

The outdoor air damper on the north schoolroom was fixed in place to provide the same percentage of fresh airflow measured for a typical room with the fixed louvers (Figure B-4).

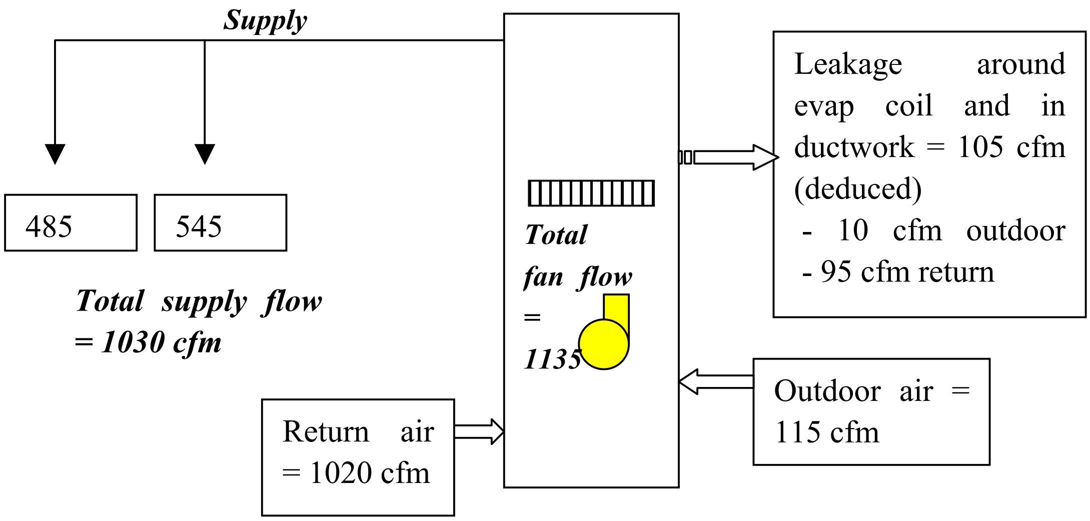


Figure B-4 - Airflow Balance for Test Room

To achieve control at this relatively low airflow, the damper was fixed at a barely open position (about 10%). The damper blade does not seal well and the required outdoor airflow could almost be obtained with the damper in a "closed" position. For this room, the measured flow rates after fixing the damper were:

- Total supply airflow at grill = 1030 cfm
- Total return airflow at grill = 1020 cfm
- Outdoor air flow into unit = 115 cfm
- Total supply fan flow (return + OA) = 1135 cfm
- Leakage in evap coil and ductwork (deduced) = 105 cfm
- Source of supply airflow (assuming complete mixing in fan)
  Return air 925 cfm
  Outdoor air = 105 cfm
  TOTAL 1030
- % Outdoor air to room = $\frac{105}{1030} = 10.2\%$

To achieve control at this relatively low airflow, the damper was fixed at a barely open position (about 10%). The damper blade does not seal well and the required outdoor airflow could almost be obtained with the damper in a fully closed position.

CO_2 Measurement Equipment

Internal CO_2 measurements were recorded at the return air grill and at two locations within the test room to measure the approximate bulk room air concentration. The return air measurement was recorded using the Virtual Mechanic monitoring system installed as part of the project, while the internal bulk room CO_2 concentrations were measured in opposite areas of the room using Micro Data Logger provided on loan by Architectural Energy Corporation. Figure B-5 shows the location of the CO_2 measurement points.

The system modifications were completed May 7, 2002 and measurements continued until the school year ended on June 7.

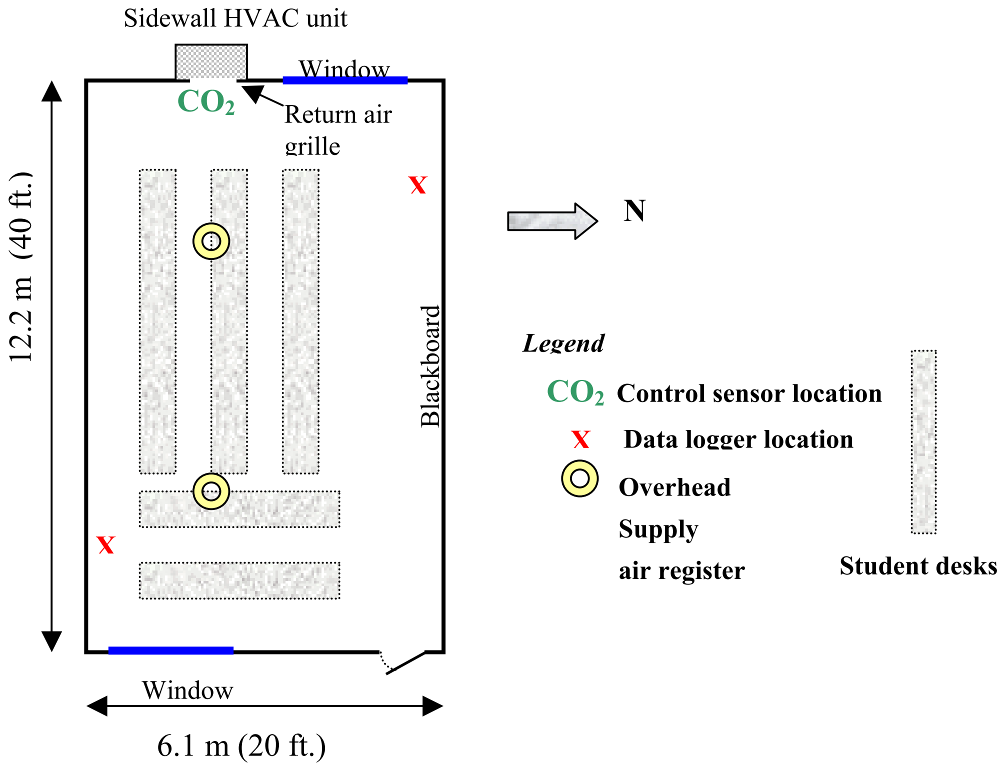


Figure B-5 - CO_2 Measurement Locations in North Test Room

Results

The average bulk room and return air CO_2 concentrations measured during this test period were calculated during school days between 8 am and 3 pm. The levels are compared from two different perspectives. First, a comparison is made of the average CO_2 values measured at the north test room with the low fixed ventilation rate with those during the same time period in the south test room that operated with the demand control ventilation system "on" and a CO_2 setpoint at 800 ppm_v. This comparison is given in Table B-1. When the DCV system is on, the minimum damper position is set to 0% and only opens when the CO_2 exceeds the setpoint or when economizer cooling mode is called for. Secondly, the test room with the minimal fixed ventilation is compared value in the same room (with the same student class) when it was set up for DCV "on" (setpoint = 800 ppm_v) and the DCV control "off". When the DCV system is off, the minimum damper position is set at 40%, which provides 450 cfm of fresh airflow (see Figure B-2) and translates to approximately 15 cfm per person. The results of this second comparison are given in Table B-2.

Table B-1 - Comparison of Internal CO_2 Concentrations at Test Room and Second Room with DCV System On

Room, Configuration	Mean Return $[CO_2]$ ppm_v	Mean Bulk Room $[CO_2]$ ppm_v	Supply Air $[CO_2]$ ppm_v	Ambient $[CO_2]$ ppm_v
North room, Fixed ventilation	1075	1218	992	401
South room, DCV on	728	790	580	401

Table B-2 - Comparison of Internal CO_2 Concentrations at Test Room and the Same Room with DCV System On

Room, Configuration	Mean Return $[CO_2]$ ppm_v	Mean Bulk Room $[CO_2]$ ppm_v	Mean Supply Air $[CO_2]$ ppm_v	Ambient $[CO_2]$ ppm_v
North room, Standard fixed louver ventilation (5/7 - 6/7/02)	1075	1218	992	401
North room, DCV on (3/13, 4/10 -4/15, 4/29-5/6)	638	708	525	399
North room, DCV off [15 cfm/person] (3/13-4/10, 4/16-4/26)	687	768	559	396

In this analysis, the supply air concentration is computed using a volume balance based on the return and ambient air CO_2 concentrations and the resulting airflows at the current damper position. For the first test case, the mean return concentration is 48% higher and the mean bulk room concentration is 54% in the test room with the air ventilation set equal to the same flow found in modular schoolrooms with the factory fixed air louvers as compared to the room that had an active DCV system.

In the second comparison, the standard fixed louver ventilation configuration without controllable dampers results in a mean return concentration 72% greater than the same room with the DCV on and 59% higher than the same room with the DCV system off and ventilation set for 15 cfm per person.

A plot, given in Figure B-6, of the bulk room CO_2 concentration in the test room with ventilation equal to the standard fixed louvers also provides insight. This plot shows the CO_2 concentrations between 8 am and 4 pm daily for a representative week during the test period.

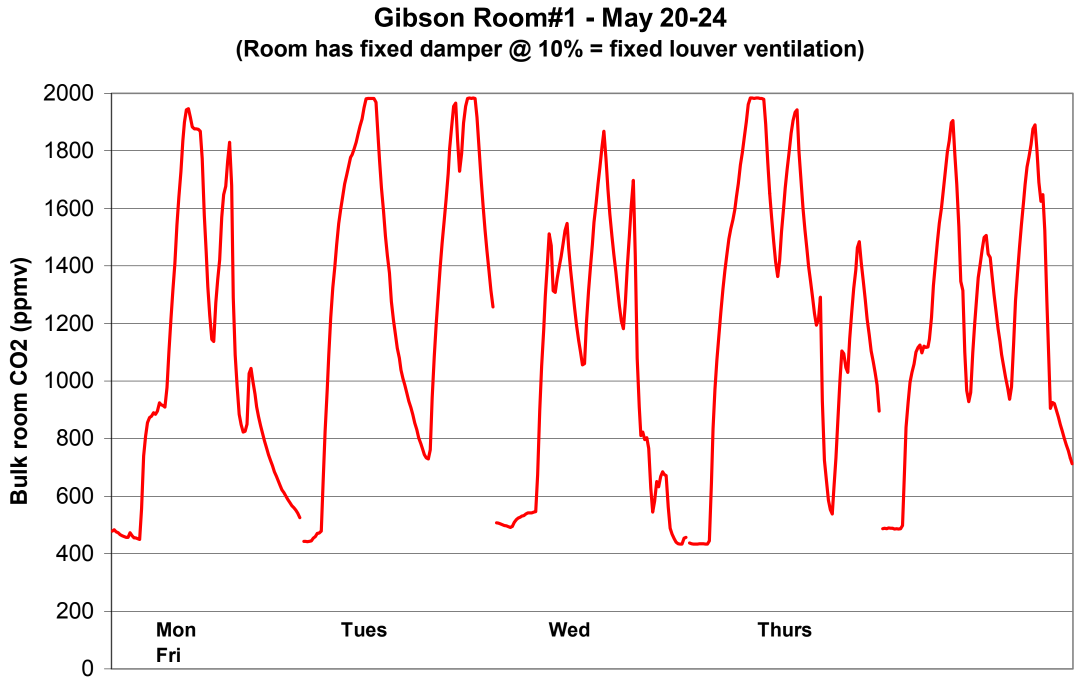


Figure B-6 - Bulk Room CO_2 in Test Room for Representative Week

Note that the bulk room CO_2 concentrations at some times approached or even perhaps exceed the 2000 ppm upper limit of the CO_2 sensor. The interior concentration remained above 1000 ppm after 4 pm, when the system fan is scheduled to shut off, for three of the days during this week.

Conclusion and Discussion

A significantly higher CO_2 concentration was measured in the test room set up for ventilation air approximating that of a standard room without controllable dampers. This indicates a poorer indoor air quality due to outdoor air ventilation rates that do not meet minimum standards set by ASHRAE or the California Title 24 requirements.

Thus, it appears that retrofitting DCV systems may have significant benefits for improving indoor air quality compared to the HVAC package with fixed ventilation louvers at a typical modular schoolroom. Such a retrofit would improve indoor air quality while still perhaps saving energy, although it is unsure whether school districts will do so due to the cost of the retrofit. The retrofit would need to include the damper assembly (approximately $250 wholesale) as well as the upgraded sensor system and labor cost. In addition, during the retrofits at the test rooms for this study, the main internal power wiring of the HVAC unit had to be rerouted in order to make room for the controllable damper assembly. This situation would increase the labor and miscellaneous materials cost for the retrofit.

NISTIR 7042

Simulations of Indoor Air Quality and Ventilation Impacts of Demand Controlled Ventilation in Commercial and Institutional Buildings

Andrew Persily
Amy Musser
Steven Emmerich
Michael Taylor

National Institute of Standards and Technology
Technology Administration, U.S. Department of Commerce

NISTIR 7042

Simulations of Indoor Air Quality and Ventilation Impacts of Demand Controlled Ventilation in Commercial and Institutional Buildings

Andrew Persily
Building and Fire Research Laboratory
Amy Musser
University of Nebraska
Steve Emmerich
Building and Fire Research Laboratory
Michael Taylor
University of Nebraska

Prepared for:
Architectural Energy Corporation
Boulder, Colorado

August 2003

U.S. Department of Commerce
Donald L. Evans, *Secretary*

Technology Administration
Phillip J. Bond, *Undersecretary of Commerce for Technology*

National Institute of Standards and Technology
Arden L. Bement, Jr., *Director*

ABSTRACT
Carbon-dioxide (CO_2) based demand controlled ventilation (DCV) offers the potential for more energy efficient building ventilation compared with constant ventilation rates based on design occupancy levels. A number of questions related to CO_2-based DCV have been raised concerning the indoor air quality impacts, primarily with respect to contaminants with source strengths that are not dependent on the number of occupants. In addition, questions exist regarding potential energy efficiency benefits, optimal control strategies for different building types, and sensor performance and deployment. In order to obtain some insight into the issue of IAQ impacts of CO_2-based DCV, a simulation study was performed in six commercial and institutional building spaces using the multizone airflow and IAQ model CONTAMW. These simulations compared seven different ventilation strategies, four of which used CO_2 DCV. The simulations, performed for six U.S. cities, were used to compare ventilation rates, indoor CO_2 levels, indoor concentrations of a generic volatile organic compound (VOC) as an indicator of non-occupant contaminant sources, and energy impacts. The results indicate that these impacts are dependent on the details of the spaces including occupancy patterns, design ventilation rate and ventilation system operating schedule, as well as the specific assumptions used in the analysis including contaminant source strengths and system-off infiltration rates. For the cases studied, the application of CO_2 DCV resulted in significant decreases in ventilation rates and energy loads accompanied by increased indoor CO_2 and VOC concentrations. The increases in CO_2 were not particularly large, in the range of 180 mg/m^3 (100 ppm(v)). The indoor VOC levels increased by a factor of two to three, but the absolute concentrations were still relatively low based on the assumed emission rates. The annual energy load reductions were significant in most of the cases, ranging from 10 % to 80 % depending on the space type, climate, occupancy schedule, and ventilation strategy.

Keywords: carbon dioxide, control, energy efficiency, indoor air quality, modeling, simulation, ventilation, volatile organic compounds

TABLE OF CONTENTS

1. INTRODUCTION AND BACKGROUND

Commercial building ventilation systems are designed, installed and operated to heat and cool occupied spaces to achieve thermal comfort and to provide outdoor air to the occupants. Outdoor air ventilation is provided to buildings primarily to dilute contaminants that are generated by building occupants and their activities and by building materials and furnishings. The rate at which outdoor air is brought into a building by its ventilation system is determined during the building design based on requirements of applicable building codes and standards. For example, ASHRAE Standard 62-2001 (ASHRAE 2001a) and California Energy Efficiency Standards, so-called Title 24, (CEC 2001) contain minimum ventilation requirements for a number of different occupancy types in units of L/s (cfm) per person and in L/s per m^2 of floor area (cfm/ft^2).

Determining design outdoor air ventilation rates for commercial buildings using Standard 62-2001 or Title 24 is a relatively straightforward process. For each space served by a given ventilation system, one determines the expected or design number of occupants for space types with ventilation requirements in units of L/s (cfm) per person. In spaces with requirements in units of $L/s\text{-}m^2$ (cfm/ft^2), one determines the floor area. Based on the ventilation requirements contained in the standard, these values (number of people and floor area) are used to determine the L/s (cfm) of outdoor air required by that space under full occupancy. Standard 62-2001 also requires that these rates be adjusted to account for ventilation effectiveness (degree of ventilation air mixing in the space). Also, if the spaces are served by a system that recirculates air from multiple spaces and redistributes it along with "new" outdoor air, Standard 62-2001 requires the use of the so-called "multiple spaces" approach to determine the outdoor air intake rate. If no such recirculation occurs, then the outdoor air intake rate is equal to the sum of the outdoor air requirements for all the spaces served by the system, after adjusting for ventilation effectiveness.

ASHRAE Standard 62-2001 also allows for a reduction in the design occupancy under conditions of intermittent occupancy. Basically, the standard allows one to use the average occupancy instead of the design occupancy for spaces where the design occupancy is based on a peak lasting 3 h or less. However, one is not permitted to reduce the design occupancy by any more than 50 %. The average occupancy is then multiplied by the per person ventilation requirement for the space to determine the design outdoor air intake rate. Note that this reduction cannot also be employed when demand controlled ventilation is also used.

Demand controlled ventilation (DCV) is a ventilation rate control strategy to address the concern that when a space is occupied at less than its design occupancy, unnecessary energy consumption can result if the space is ventilated at the design minimum rate rather than the ventilation rate based on the actual occupancy. Furthermore, early during a given day of building occupancy, contaminants generated by people and their activities will not yet have reached their ultimate levels based on the transient nature of contaminant buildup. As a result, it is sometimes possible to delay or lag the onset of the design ventilation rate to take credit for this transient effect. A number of approaches have been proposed to account for actual occupancy levels and to provide the ventilation rate corresponding to actual rather than design occupancy. These include time-based scheduling when occupancy patterns are predictable, occupancy sensors to determine when people have entered a space (though not necessarily how many), and carbon dioxide (CO_2) sensing and control as a means of estimating the number of people in a space or the strength of occupant-related contaminant sources.

Controlling outdoor air intake rates using CO_2 DCV offers the possibility of reducing the energy penalty of over-ventilation during periods of low occupancy, while still ensuring adequate levels of outdoor air ventilation. As discussed later in this report, depending on climate and occupancy patterns, CO_2 DCV may provide significant energy savings in commercial and institutional buildings. While a number of studies have suggested the extent of such savings via field studies and computer simulations, additional work is needed to better define the magnitude of energy savings possible and the dependence of these savings on climate, building and system type, control approach, and occupancy patterns. In addition, important issues remain to be resolved in the application of CO_2 DCV including how best to apply the approach, which in turn includes issues such as which control algorithm to use in a given building, sensor location, sensor maintenance and calibration, and the amount of baseline ventilation required to control contaminant sources that don't depend on the number of occupants.

An earlier report presented a state-of-the-art review of CO_2 DCV technology and its application in commercial and institutional buildings (Emmerich and Persily 2001). That report presented discussions of CO_2 generation rates by people, the relationship of indoor CO_2 to building ventilation rates, and the basic concept of controlling ventilation based on indoor CO_2 levels. It also contained a literature review of previous research on CO_2 DCV, including field demonstration projects, computer simulation studies, studies of sensor performance and location, and discussions of the application of the approach. This earlier report and other discussions of CO_2 DCV identified indoor air quality impacts as an important issue in the application and performance of these systems. The key indoor air quality concern relates to contaminants that are generated in a building at a rate that does not depend on the number of occupants. For example, building materials and furnishings emit contaminants at an approximately constant rate independent of the occupancy level, including when the building is empty. Questions have been raised as to how well these contaminants will be controlled by a DCV system when the occupancy level is low. Some have proposed maintaining a minimum outdoor air ventilation rate at all times to control these contaminants, with the minimum based on a specific outdoor air intake rate per unit floor area expressed in L/s•m^2 (cfm/ft^2) (CEC 2001) or as a fraction of the design outdoor air intake rate, for example 25 % (Schell et al. 1998).

ASHRAE Standard 62-2001 allows for the outdoor air intake rate to be adjusted based on variations in occupancy (as noted earlier in the discussion of the intermittent occupancy approach), but regardless of the approach used to make these adjustments the system must still provide the required outdoor air ventilation rate per person. The standard does not explicitly discuss CO_2-based DCV in terms of sensor location, minimum outdoor airflow rates or other details. However, a number of official interpretations to the standard issued by ASHRAE make it clear that these approaches can comply with the standard if properly implemented. Title 24 also allows the use of demand controlled ventilation. If fact, it is required in spaces with high occupant densities as an energy efficiency measure.

As noted above, the outdoor air ventilation requirements in ASHRAE Standard 62-2001 are largely expressed as airflow rates per person in L/s•person (cfm/person). In some spaces, for example corridors and retail spaces, they are expressed in L/s•m^2 (cfm/ft^2) of floor area. The per person requirements are intended to address contaminants emitted by the occupants themselves as well as by the space they occupy, including building materials, furnishings and equipment. In developing the ventilation requirements per person, there is an implicit assumption as to the number of occupants per unit floor area in order to handle these non-occupant contaminants. If the space being designed has a different occupant density, it may receive more or less outdoor air than needed to handle the floor-area contaminants. In order to address that concern, a revision of the Ventilation Rate Procedure in ASHRAE Standard 62 has been developed that contains per person and per floor area outdoor air requirements for all spaces (Persily 2001). Under the revision, referred to as addendum 62n, one multiplies the number of people in a space by a per person ventilation requirement R_p and multiplies the floor area of the space by a per floor area requirement R_a. These two products are then added together to determine the outdoor air requirement in the occupied zone of the space. Further adjustments are required to account for mixing in the space and system effects in recirculating systems serving multiple spaces. This so-called additive approach has the advantage of addressing the concern about non-occupant contaminant sources and the provision of ventilation to handle these sources when occupancy is low or zero. It could also make the application of CO_2 DCV more challenging compared with ventilation requirements expressed solely in terms of per person rates, but control algorithms have been developed to implement CO_2 DCV for so-called "additive" ventilation requirements (Sowa 2002).

Resolving all the issues related to the application of CO_2 DCV in commercial buildings, including the energy and IAQ impacts, will require field testing and application experience, as well as simulation studies. A number of modeling studies have looked at energy impacts of CO_2 DCV strategies in different building types and different climates (e.g., Brandemuehl and Braun 1999). Other simulation studies have focused on the indoor air quality implications of CO_2 DCV (e.g., Carpenter 1996, Emmerich et al. 1994, Enermodal 1995). The study described in this paper employs an airflow and contaminant dispersal model to investigate the issue of how CO_2 DCV and other ventilation strategies impact indoor air quality and ventilation. In particular, the simulations are focused on how CO_2 DCV impacts the control of non-occupant contaminants, in this case a generic volatile organic compound (VOC), generated at a constant rate to represent contaminants emitted by building materials and furnishings.

These simulations are performed using the airflow and indoor air quality model CONTAMW (Dols and Walton 2002) for six commercial and institutional building spaces. The results are then used to compare ventilation rates, contaminant concentrations and energy associated with ventilation for seven ventilation strategies: constant ventilation volumes specified in ASHRAE Standard 62-2001 and addendum 62n, a theoretical demand control strategy that perfectly tracks occupancy, and four CO_2 DCV strategies with different maximum and minimum flow rates, including one based on California's Title 24 requirements.

2. DESCRIPTION OF ANALYSIS

The simulations in this study were performed using the multizone network airflow and contaminant dispersal model CONTAMW (Dols and Walton 2002). This model allows one to represent a building as a collection of interconnected zones and then calculates airflow rates induced by weather and ventilation system operation based on air leakage characteristics of the boundaries between zones and pressures and on ventilation system airflows. The user can also enter contaminant source strengths to calculate concentrations over time based on the calculated airflow rates and other mass transport mechanisms (e.g. filtration, deposition, chemical reaction). The latest version of CONTAMW can simulate the performance of controls, in which an airflow rate, fan or damper is controlled based on the contaminant concentration, temperature or pressure in a zone. The simulations described in this report employed this new capability in simple models of the six study spaces as a means of simulating CO_2 DCV.

2.1 Study Spaces

Building models were created in CONTAMW for six different space types. Four of these were generic spaces devised for the purposes of this study: office, conference room, lecture hall and classroom. The two other spaces were based on actual buildings being monitored as part of a larger study on CO_2 DCV being conducted as part of the same CEC-sponsored program that supported this work: portable classroom and playroom in a fast food restaurant. Table 1 summarizes the characteristics of the six spaces including floor area, ceiling height and design occupancy. For the first four spaces, the design occupancy is based on the default values given in ASHRAE Standard 62-2001 (ASHRAE 2001a). Actual dimensions of the two monitored spaces were used to construct models of the portable classroom and fast food playroom, and the occupancies were estimated based on available design information.

Space type	**Floor area m^2 (ft^2)**	**Ceiling height m (ft)**	**Design occupancy # of people**	**Occupant density #/100 m^2 (#/1000 ft^2)**	**Ventilation system operating time**
Office	1000 (10760)	3.0 (9.8)	70	7.0 (6.5)	0600-1900
Conference Room	100 (1076)	3.0 (9.8)	50	50.0 (46.5)	0600-1800
Lecture Hall	100 (1076)	6.0 (19.7)	150	150.0 (139.4)	0800-2100
Classroom	100 (1076)	3.0 (9.8)	35	35.0 (32.5)	0600-1800
Portable classroom	89 (958)	2.6 (8.5)	20	22.5 (20.9)	0700-1700
Fast food restaurant	125 (1346)	5.4 (17.7)	70	56.0 (52.0)	0600-2400

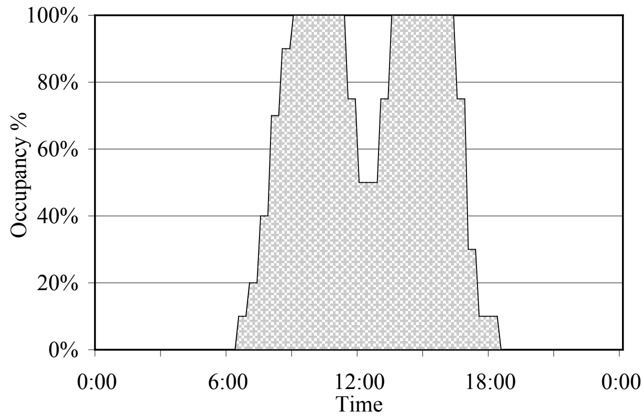
Table 1 Space Characteristics

Each space was modeled as a single zone with a ventilation system that provides outdoor air at a rate determined by the control strategy of interest, as outlined below. Details of the ventilation system equipment were not considered in this study, though they can be important; only the outdoor air intake rate is accounted for in the modeled ventilation systems. The systems are assumed to operate during the times indicated in the last column of Table 1 and to be off at night and during unoccupied periods over the weekends. A constant infiltration rate of 0.1 air changes per hour is assumed to exist in each space at all times, including when the ventilation system is operating. This value was chosen to represent a low infiltration condition that might exist under low wind speeds and moderate outdoor temperatures and result in significant buildup of contaminants before the system is activated.

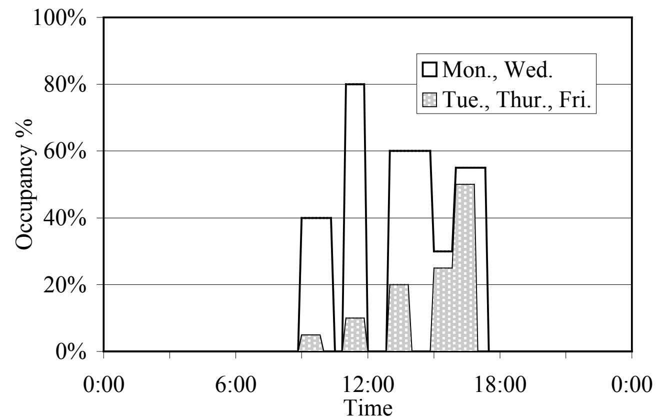
Occupancy profiles

Weekly occupancy schedules for each space are shown in Figure 1. Schedules for the four generic spaces (1a through 1d) were selected to represent realistic usage and to include scenarios that were significantly different from one another in order to test of each control scheme. The Office and Classroom tend to experience long periods at close to their design occupancies. Occupancy changes in the morning and evening are more gradual for the Office, where workers tend to arrive and depart at different times. The Classroom is more densely occupied than the Office, and operates on a more rigid schedule, with the students arriving, departing, and taking lunch at the same time. However, it was assumed that a teacher would arrive early and stay later than the students. In contrast, the Lecture Hall and Conference Room are intermittently occupied. The Lecture Hall schedule is the busier of the two, with more occupied hours in the day, and usually with 50 % or more of the design capacity when the room was occupied. The Conference Room is modeled with two occupancy profiles, with a busier schedule specified for Mondays and Wednesdays. All four of these spaces are assumed to be unoccupied on weekends, and all occupants for these spaces are specified as adults.

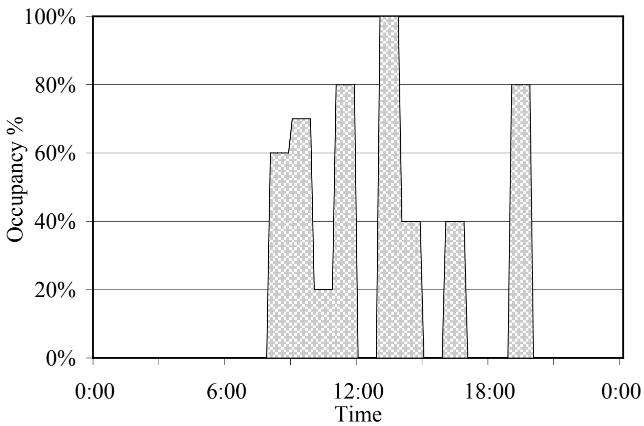
The Portable Classroom was modeled with two adults and eighteen children. The occupancy profile specified for the Classroom is also used here for the children, with different CO_2 generation rates for the adults and children based on body size. The occupancy profiles used for the Fast Food Restaurant simulations are based on actual occupancy data collected in the monitored restaurant. These data were used to develop an occupancy schedule between 0600 to 2400 seven days a week, with different schedules for weekdays and weekends.



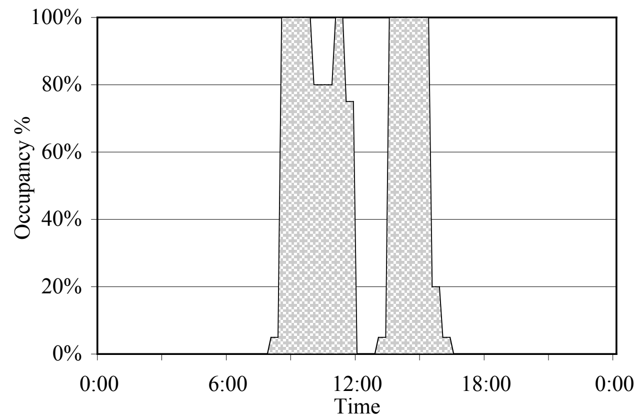
(a) Office



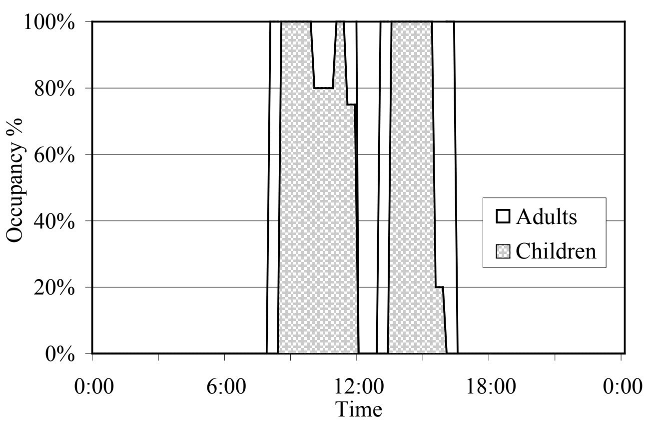
(b) Conference Room



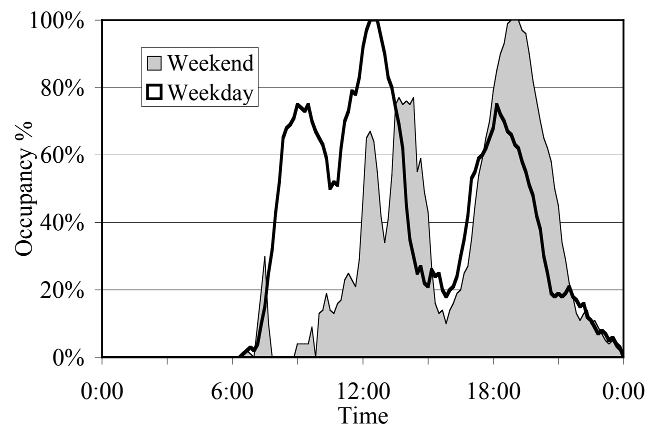
(c) Lecture Hall



(d) Classroom



(e) Portable Classroom



(f) Fast food restaurant

Figure 1 Occupancy Schedules

Contaminant generation rates

The simulations accounted for two contaminants, occupant-generated carbon dioxide (CO_2) and a generic volatile organic compound (VOC) intended to represent contaminants from building materials and furnishings. While VOC emissions in buildings are far more complex than the simple approach used here (Levin 1995, Wolkoff 1995), the objective in including VOCs in these simulations was to capture the impact of DCV systems on non-occupant sources that are relatively constant over and time by including a source strength related to the floor area of the space.

In these simulations, CO_2 was generated by adults at a rate of 0.3 L/min•person, which corresponds to an activity level consistent with office work (ASHRAE 2001b). In the Portable Classroom, children were assumed to generate CO_2 at a rate of 0.18 L/min•person, a value based on the body size of a ten year old child. Carbon dioxide generation in the Fast Food Restaurant playroom was modeled as 0.3 L/min•person, a value appropriate for both sedentary adults and small school children at an activity level of 2.5 met (ASTM 2002, EPA 1999). The emission rate for the generic VOC was assumed to be constant at a rate of 0.25 mg/h per m^2 of floor area during unoccupied periods and 0.50 mg/h•m^2 during occupancy. These values are based on limited field measurements of VOC emission rates (Levin 1995). Although actual contaminant generation rates may differ significantly for different building types, there is not sufficient data available to justify varying these rates in this study. Sorption and re-emission of VOCs from surfaces were not modeled in these simulations, and the outdoor concentrations of CO_2 and VOC were assumed to equal 720 mg/m^3 (400 ppm(v)) and 0 mg/m^3 respectively over the entire simulation period.

2.2 Ventilation Rates and Control Approaches

The ventilation rates in the spaces were based on ASHRAE Standard 62-2001, the revision to those rates described earlier (Addendum 62n), and the requirements in California's Title 24 (CEC 2001). Table 2 presents the outdoor air requirements for each space based on 62-2001, addendum 62n, and Title 24. For Standard 62-2001, the outdoor air requirements are presented in L/s•person (cfm/person) followed by the outdoor air intake requirement for the space based on the number of occupants (see Table 1). These outdoor air intake rates for each space are presented in L/s (cfm) and in L/s•m^2 (cfm/ft^2) of floor area. For addendum 62n, the outdoor air requirements are presented as both the people and area rate, which are then combined based on the number of occupants and the floor area of the space given in Table 1. Title 24 requires 7.1 L/s (15 cfm) per person based on the larger of the actual design occupancy or 50% of the exiting density specified by the Uniform Building Code (UBC), with a minimum ventilation rate of 0.76 L/s•m^2 (0.15 cfm/ft^2). Under Title 24, carbon dioxide DCV cannot be used in the Office, since the 0.76 L/s•m^2 (0.15 cfm/ft^2) minimum is larger than 7.1 L/s (15 cfm) for the assumed occupant density. Therefore the outdoor air intake for the Office is based on the floor area rather than the number of people. Also, the assumed occupancy for the Portable Classroom (20 people) is less than 50% of the UBC exiting density, so the maximum flow rate for that case is based on 24 occupants. Note that since the simulations were originally performed using IP units, the ventilation requirements in Table 2 are correct for IP units. The SI values are converted from the IP values, resulting in slight differences relative to the SI values contained in Standard 62-2001, Addendum 62n and Title 24.

For each space type, Table 2 also contains the steady-state CO_2 and VOC concentrations corresponding to the design outdoor air intake rate based on the assumed VOC and CO_2 generation rates. Note that for the Standard 62-2001 ventilation rates, the steady-state CO_2

concentrations range from about 1500 mg/m^3 (900 ppm(v)) to 1900 mg/m^3 (1100 ppm(v)), except in the two cases employing the intermittent occupancy provision of the standard. The VOC concentrations vary more widely, over a range of twenty to one, with the variation due primarily to the variation in the floor area per occupant among the spaces. The VOC levels are all range from less than 0.1 mg/m^3 to 0.2 mg/m^3, with the lowest concentrations in the more densely occupied spaces. Note that these concentrations are on the low end of those reported from field measurements in commercial buildings, which are in this range and even higher in non-problem buildings (Anderson et al. 1997, Brown et al. 1994, Daisey et al. 1994, Hadwen et al. 1997, Wolkoff 1995). Note that the concentrations in Table 2 are all steady-state values, i.e., the values that would eventually exist if the emission rate and ventilation rate were maintained long enough for steady-state conditions to be achieved. However, depending on the occupancy and ventilation schedules, steady-state conditions will not necessarily occur in these spaces.

The outdoor air ventilation rates for the six spaces based on Addendum 62n tend to be lower than those based on 62-2001, particularly in the more densely occupied spaces (Conference Room, Lecture Hall and Restaurant). In fact, part of the reason for the changes in this addendum was the concern that the existing rates in the standard were larger than necessary in densely occupied spaces due to “overcounting” of emissions from floor area-related contaminants. The ventilation rate in the Portable Classroom is slightly higher under 62n based on it having a lower occupancy density than assumed in Standard 62-2001. The lower ventilation rates under the addendum generally result in higher steady-state CO_2 levels, particularly in the densely occupied spaces. Most of the values are still in the range of 1800 mg/m^3 (1000 ppm(v)), but the two most densely occupied spaces (Conference Room and Lecture Hall) are closer to 3600 mg/m^3 (2000 ppm(v)). The VOC levels for the 62n rates are generally higher than those based on 62-2001, with the largest increases again seen in the densely occupied spaces. However, the VOC levels are still range from less than 0.1 mg/m^3 to 0.3 mg/m^3, again on the low end of concentrations measured in the field. Again, these are all steady-state concentrations, which may not necessarily occur in these spaces given the occupancy schedules. Also, these steady-state VOC concentrations are based on the ventilation rates and source strengths assumed to exist during occupied periods and neglect any impacts of higher concentrations that might occur overnight when the system is off.

For Title 24 the steady-state CO_2 levels are all approximately 2000 mg/m^3 (1100 ppm(v)), with lower levels in the Office and the Portable Classroom. In the office, the lower concentration results from its ventilation requirement under CO_2 DCV being based on 0.76 L/s•m^2 (0.15 cfm/ft^2), which is higher than 7.1 L/s (15 cfm) per person. The Portable Classroom outdoor air intake rate is based on 24 occupants rather than the actual 20 occupants based on the Title 24 requirement to assume no less than 50 % of the UBC exiting density. The steady-state VOC concentrations are similar to those seen for Standard 62-2001 and Addendum 62n.

Standard 62-2001					
		Outdoor air intake		Steady-state concentration	
Space type	Outdoor air requirement L/s (cfm) per person	L/s (cfm)	L/s•m^2 (cfm/ft^2)	CO_2 mg/m^3 (ppm(v))	VOC mg/m^3
Office	9.4 (20)	661 (1400)	0.7 (0.13)	1674 (930)	0.21
Conference Room	9.4 (20)	472 (1000)	4.7 (0.93)	1674 (930)	0.03
Lecture Hall	7.1 (15)	1062 (2250)	10.6 (2.09)	1991 (1106)	0.01
Classroom	7.1 (15)	248 (525)	2.5 (0.49)	1991 (1106)	0.06
Portable classroom	7.1 (15)	142 (300)	1.6 (0.31)	1532 (851)	0.09
Fast food restaurant	9.4 (20)	661(1400)	5.3 (1.04)	1674 (930)	0.03
Conference room*	9.4 (20)	236 (500)	2.4 (0.47)	2626 (1459)	0.06
Lecture Hall*	7.1 (15)	531 (1125)	5.3 (1.05)	3262 (1812)	0.03
* Under intermittent occupancy provision of Standard 62-2001					
Addendum 62n					
		Outdoor air intake		Steady-state concentration	
Space type	Outdoor air requirement L/s (cfm) per person/ L/s•m^2 (cfm/ft^2)	L/s (cfm)	L/s•m^2 (cfm/ft^2)	CO_2 mg/m^3 (ppm(v))	VOC mg/m^3
Office	2.4/0.3 (5.0/0.06)	470 (996)	0.5 (0.09)	2061 (1145)	0.30
Conference Room	2.4/0.3 (5.0/0.06)	149 (315)	1.5 (0.29)	3740 (2078)	0.09
Lecture Hall	3.5/0.3 (7.5/0.06)	562 (1190)	5.6 (1.11)	3123 (1735)	0.03
Classroom	4.7/0.6 (10/0.12)	226 (479)	2.2 (0.44)	2113 (1174)	0.06
Portable classroom	4.7/0.6 (10/0.12)	149 (315)	1.7 (0.33)	1494 (830)	0.08
Fast food restaurant	3.5/0.9 (7.5/0.18)	362 (767)	2.9 (0.57)	2461 (1367)	0.05
Title 24					
		Outdoor air intake		Steady-state concentration	
Space type	Outdoor air requirement	L/s (cfm)	L/s•m^2 (cfm/ft^2)	CO_2 mg/m^3 (ppm(v))	VOC mg/m^3
Office	0.76 L/s•m^2 (0.15 cfm/ft^2)	762 (1614)	0.75 (0.15)	1546 (859)	0.18
Conference Room	7.1 L/s (15 cfm) per person	354 (750)	3.5 (0.70)	1991 (1106)	0.04
Lecture Hall	7.1 L/s (15 cfm) per person	1062 (2250)	10.6 (2.09)	1991 (1106)	0.01
Classroom	7.1 L/s (15 cfm) per person	248 (525)	2.5 (0.49)	1991 (1106)	0.06
Portable classroom	7.1 L/s (15 cfm) per person	170 (360)	1.9 (0.38)	1399 (777)	0.07
Fast food restaurant	7.1 L/s (15 cfm) per person	495 (1050)	4.0 (0.78)	1991 (1106)	0.04

Table 2 Design Ventilation Rates and Steady-State Contaminant Levels during Occupancy

Seven ventilation control scenarios were simulated in the six spaces, with one additional scenario applied to two of them. The first three scenarios, based on ASHRAE Standard 62, serve as reference cases for comparing the DCV options. They are:

62/2001: Constant outdoor air intake rates based on ASHRAE Standard 62-2001 and the design occupancy values in Table 1.

62tracking: Outdoor air intake rates that track occupancy (as depicted in Figure 1) perfectly using the ASHRAE Standard 62-2001 rates, i.e., the intake rate always equals the number of occupants times the per person ventilation requirement.

62/Int: Outdoor air intake rate based on 50 % of peak occupancy using the intermittent occupancy approach in the standard (only applied to the Conference Room and Lecture Hall).

Two cases employing CO_2 DCV using the Standard 62-2001 rates were also studied:

C-ZeroMin: Maximum ventilation rate based on ASHRAE Standard 62-2001; minimum ventilation rate equal to zero.

C-25%Min: Maximum ventilation rate based on ASHRAE Standard 62-2001; minimum ventilation rate equal to 25 % of the maximum.

In addition, two cases were studied based on the revision of the Ventilation Rate Procedure in the standard, so-called addendum 62n:

62n: Constant outdoor air intake rates based on addendum 62n and the design occupancy values from Table 1.

C-62nAreaMin: CO_2 DCV control with the maximum ventilation rate based on the design occupancy and the requirements in addendum 62n; minimum ventilation rate equal to the "area" requirement times the floor area of the space.

Finally, one case followed the requirements of California's Title 24:

C-T24: CO_2 DCV control with the maximum ventilation rate based on the requirement for 7.1 L/s (15 cfm) person in these spaces, using the larger of the design occupancy or 50 % of the UBC exiting density. The minimum ventilation rate is based on 0.76 L/s•m^2 (0.15 cfm/ft^2). In the case of the office, the ventilation rate is constant at this minimum level because the per person requirement results in a ventilation rate that is lower than this value.

For the reference cases and the 62n case, it was not necessary to model outdoor air intake controls. In the simulations, the ventilation system was simply scheduled to turn on and off per the operating schedules described earlier. In the case of 62tracking, in which the outdoor air intake rate was "controlled" to track occupancy perfectly, the fan was set to follow the same schedule as the occupancy. For the cases in which CO_2 control was implemented, the control simulation capabilities of CONTAMW were employed. A proportional control algorithm based on previously published descriptions was used (Schell, et al. 1998, Schell and Int-Hout 2001). The proportional controllers were specified to modulate the ventilation rate between the minimum and maximum rates with a linear response to CO_2 concentration based on the output *O* described below. The lower limit of this range was selected to be 90 mg/m^3 (50 ppm(v)) higher than the outdoor level, and the upper CO_2 limit was set at the equilibrium concentration corresponding to the design occupancy and design ventilation rate under steady conditions C_{eq}. For the Title 24 case, the upper CO_2 limit was set to 1440 mg/m^3 (800 ppm(v)) for all cases, with constant flow at the maximum ventilation rate delivered at all concentrations above this level. The CONTAMW proportional control algorithm calculates the output *O* according to the following relationship,

$$O = I \, x \, K_p \tag{1}$$

where I is the controller input and K_p is a constant. In this control strategy the controller input is the indoor CO_2 concentration minus 810 mg/m^3 (450 ppm(v)), and K_p defined as,

$$K_p = 1 / [C_{eq} - 450 \ ppm(v)] \tag{2}$$

Other control algorithms have been proposed and employed for CO_2 DCV, such as proportional-integral control.

2.3 Airflow and contaminant analysis

Each of the cases was simulated in the six spaces for a period of 7 days. The simulations were performed using a 5 min time step and yielded a CO_2 and VOC concentration at each time step. In the case of the DCV systems, the simulations also yielded a ventilation rate. These ventilation data were analyzed to yield the average ventilation rate during the occupancy period. The CO_2 concentration data were analyzed to yield the average concentration over the occupancy period and the peak hourly average during occupancy. The VOC data were also analyzed to determine the average concentration during occupancy, plus the peak concentration. Plots of CO_2 and VOC concentrations during the simulation period are presented in the results section.

2.4 Energy analysis

In order to compare the energy consumption associated with the different ventilation control cases, a simplified approach was used to estimate the heating and cooling loads associated with conditioning the ventilation air to the indoor conditions based on the sensible and latent heat capacity of the outdoor air relative to the indoor air. Therefore, the energy analysis accounts for only the load due to ventilation air, and not the energy required to meet that load, which depends on the type of system used to meet that load. Economizer operation is, however, taken into account by not including any cooling energy consumed when operating in this mode. Also, no energy consumption is assessed when the outdoor air temperature is between the heating balance point and the space temperature

<u>Determination of heating balance point temperature</u>

The balance point temperature was estimated using a simplified steady-state energy balance in which the heat transferred out of the structure equals the heat transferred in via airflows and internal gains:

$$(\Sigma UA + QC_p)\, T_o + q = (\Sigma UA + QC_p)\, T_i \tag{3}$$

where,

ΣUA = building envelope thermal conductance
Q = mass flow rate of ventilation air
C_p = specific heat of air
T_o = outdoor temperature
T_i = indoor temperature
q = internal heat gains

The heating balance point temperature is the outdoor temperature at which internal heat gains are equal to the heat loss rate at a given ventilation rate and indoor temperature, T_i. Heating is required below this temperature, and internal gains maintain the building at the heating setpoint above this temperature. The heating balance point, T_{hbp}, can be defined as:

$$T_{hbp} = T_i - q / (\Sigma UA + QC_p) \tag{4}$$

The thermal conductance term (ΣUA) is often much smaller than the ventilation flow term in equation (4) for commercial buildings, and is sometimes neglected. However, since some of the airflow control strategies allowed the airflow to occasionally go to zero, it was maintained for this analysis. However, the accuracy of the thermal conductance term is not critical, since the QC_p term is usually much larger than the UA term. The value of ΣUA was estimated based on

ASHRAE Standard 90.1 envelope requirements (ANSI/ASHRAE/IESNA 1999). During the few times when the ventilation flow rate does approach zero, the heating balance point is so low that it was not reached in any of the climates investigated.

Table 3 shows the internal heat gain used for each space, which were estimated using data published for nonresidential cooling and heating load calculations by ASHRAE (2001b). Occupant heat gains were based on the average modeled occupancy during the occupied period and assumed occupant activity levels. Heat gain from lighting was estimated using the installed lighting load from ASHRAE Standard 90.1 (ANSI/ASHRAE/IESNA 1999), adjusted for usage and allowance factors. Heat gain from 80 computers was included for the office.

	Occupants	**Lighting**	**Computers**	**Total**
Space	W/m^2	W/m^2	W/m^2	W/m^2
Office	4.9	12.1	10.0	27.0
Conference Room	9.8	6.4	0.0	16.2
Lecture Hall	42.0	9.3	0.0	51.3
Classroom	16.1	9.3	0.0	25.4
Portable Classroom	7.8	9.3	0.0	17.1
Fast Food Restaurant	17.3	20.1	0.0	37.3

Table 3 Internal Heat Gains Used to Estimate Balance Point

The weekly CONTAM simulations, repeated for an entire year, were used to determine the ventilation mass flow rate in equation (4). And because the heating balance point depends on this flow rate, it was calculated individually for each hour of the year.

Heating load

Based on the estimated heating balance point temperature, the heating load associated with ventilation was calculated for each hour in which the outdoor temperature was below the heating balance point using the relationship:

$$q_{heating} = QC_p\ (T_i - T_o) \tag{5}$$

where,

$q_{heating}$ = heating load
Q = mass flow rate of ventilation air
C_p = specific heat of air
T_o = outdoor temperature
T_i = indoor temperature (assumed 22 ºC year round)

Cooling load

When the outdoor temperature is greater than the indoor temperature, the additional sensible cooling load $q_{cooling}$ associated with the ventilation air can be calculated from the relationship:

$$q_{cooling} = QC_p\ (T_o - T_i) \tag{6}$$

Whether or not a latent cooling load exists depends, to some extent, on the latent loads as well as the degree of humidity control that can be achieved by the means of thermal conditioning in the space. For this simplified model, it was assumed that the thermal control strategy was capable of maintaining a maximum indoor relative humidity of 60 %. Therefore, when the outdoor humidity ratio exceeds this humidity ratio upper limit, a latent load associated with ventilation is assessed:

$$q_{latent} = Q\ h_{fg}\ (W_o - W_{limit}) \tag{7}$$

where,

q_{latent} = latent cooling load
Q = mass flow rate of ventilation air
h_{fg} = latent heat capacity of moist air
W_o = outdoor humidity ratio
W_{limit} = indoor humidity ratio (defined at 60 % relative humidity at 22 ºC)

The energy calculations assume that each space operates with a return air temperature economizer that uses outdoor air for cooling. Under this strategy the system provides 100 % outdoor air when the outdoor dry bulb temperature is between the supply air temperature and the indoor temperature. Some mechanical cooling will be required under these conditions, but it will be less than would be needed if indoor air was recirculated. When the outdoor temperature is above the heating balance point but below the system supply temperature, the economizer strategy mixes return air with a volume of outdoor air greater than that needed for ventilation, and neither cooling nor heating is required. In both of these cases, thermal conditioning and control, not the ventilation control strategy, dictate the amount of outdoor air supplied to the spaces. Since our approach is intended to determine the heating and cooling loads associated with ventilation, cooling energy consumed in this mode is not included in the reported values.

The heating, cooling, and latent loads associated with ventilation were calculated for every hour during which the ventilation system was assumed to be operating. These were then summed for each case over an entire year of weather data for the six cities identified below.

Climates analyzed

Based on the methodology outlined above, the energy consumption was estimated for four California climates (Bakersfield, Los Angeles, Sacramento, and San Francisco) selected to cover a range of coastal and inland climates. As points of reference, Miami (hot and humid) and Minneapolis (cold) were also analyzed. These energy estimates employed TMY2 weather data (Marion and Urban 1995), except for Sacramento and Miami for which WYEC data (ASHRAE 1997) was used. Table 3 summarizes the weather data for these six climates.

City	Heating degree days °C (°F)	Cooling degree days °C (°F)
Bakersfield	1213 (2183)	1210 (2178)
Los Angeles	1010 (1816)	341 (614)
Sacramento	1579 (2842)	643 (1157)
San Francisco	1690 (3042)	60 (108)
Miami	114 (205)	2243 (4037)
Minneapolis	4532 (8158)	325 (585)

Table 4 Summary of Six Climates Analyzed (Knapp et al. 1980)

3. RESULTS

This section presents the results of the simulations for the six space types, seven ventilation control approaches and six climates. These results are presented separately for the ventilation rates, CO_2 and VOC contaminant concentrations, and energy loads.

3.1 Ventilation Rates

The ventilation rates for the different space types and control strategies are summarized in Table 5. Note that these rates are inputs to the contaminant simulations for three of the cases (62/2001, 62 tracking and 62n), as well as 62/Int when relevant, but are calculated during the simulations for the four DCV cases (C-ZeroMin, C-25%Min, C-62nMinArea and C-T24). Also note that the intermittent occupancy case is only applied to the Conference Room and Lecture Hall. Also, while the Title 24 case is thought of as DCV, in the case of the Office it is in fact a constant ventilation rate based on the minimum outdoor air requirement of 0.76 L/s•m^2 (0.15 cfm/ft^2).

For each space and control strategy, Table 5 contains the average, minimum and maximum outdoor air intake during occupancy in units of airflow rate per person L/s•person (cfm/person). Also, in the first column, the table presents the per person design value for outdoor air intake for Standard 62-2001, addendum 62n and Title 24. The calculations were initially performed in IP units and converted to SI units. Therefore, some of the SI values are slightly different from those that appear in Standard 62, addendum 62n and Title 24. The last column of the table contains the minimum and maximum per person outdoor air intake during periods of time when the space is at its maximum occupancy level. For some spaces (e.g., Office) the space is at maximum occupancy for many hours, while for other spaces (e.g., Fast Food) maximum occupancy occurs for only short periods of time. Note that the Conference Room is never at its design occupancy of 50 people, but rather has a maximum occupancy of 40.

The maximum rates during occupancy in Table 5, for all but the 62tracking case, are well above the relevant requirements of Standard 62, 62n or Title24. This is particularly true for cases 62/2001 and 62n in which the design outdoor air intake is in effect whenever the system operates, which results in high per person rates when the occupancy is low. For the 62tracking case, the averages, minimums and maximums are all equal to the Standard 62 requirement as expected, except in the Portable Classroom because the 62tracking case is actually based on the CO_2 generation rates of the occupants. (Since different CO_2 generation rates are used for the adults and children in that space, there is some variation in the per person rates based on whether the space is occupied by only the adults or by the whole class.) The minimum per person rate for the case of C-ZeroMin is always zero, since the intake is at its minimum position (zero intake) until the CO_2 levels build-up after the space has been occupied for some time. The other CO_2 DCV cases also have minimum per person rates below the rate required by the standard, as the indoor CO_2 levels are too low at the start of occupancy for the CO_2 controls to induce outdoor air intake. However, these low rates are temporary and not unexpected.

Figures 2 through 7 are plots of the total ventilation rates, including infiltration, for each of the spaces and ventilation control approaches over a period of one or two days. The one-day plots all contain data for a Friday, to capture the impact of any CO_2 buildup over the week in cases where the assumed infiltration rate of 0.1 h^{-1} is not sufficient to bring the indoor CO_2 level down to the outdoor level overnight. For the CO_2 control cases, this residual CO_2 leads to a low level of ventilation early in the morning. For the spaces with occupancy patterns that vary by day of the week, two days are presented in the corresponding figure. Specifically, Figure 3 presents the ventilation rates for Wednesday and Thursday in the Conference Room, and Figure 6 presents

Friday and Saturday for the Fast Food Restaurant. In all the figures, the 62-2001 and 62n cases are horizontal lines indicating constant ventilation rates when the system operates. The Title 24 case C-T24 results in a constant intake rate for the office space as discussed earlier. The intermittent occupancy cases (62-Int) in the Conference Room and the Lecture Hall, Figures 3 and 4 respectively, also exhibit constant ventilation rates. The 62tracking case appears as a solid black line in all the figures, with the ventilation rate corresponding to the occupancy schedule of the given space. The four control cases (C-ZeroMin, C25%Min, C-62nAreaMin and C-T24) exhibit more variation as the controls respond to the indoor CO_2 level. They generally start each day low relative to the constant ventilation rate cases, with the ventilation rates increasing as the indoor CO_2 levels increase.

Referring to Table 5, the results for the Office exhibit a number of trends that are also reflected in most of the other spaces. The average per-person outdoor air intake rate is the second highest for the 62/2001 case in which the intake rate is always equal to 9.4 L/s (20 cfm) times the maximum number of occupants. Under low occupancy, the constant intake rate is divided by a relatively small number of people, yielding per person ventilation rates with a maximum value of 94 L/s (200 cfm). The lowest average intake rate is for 62tracking, in which the system always brings in 9.4 L/s (20 cfm) per person times the number of people in the space. The 62n rate is also constant during system operation, but at a lower value than 62/2001, resulting in a lower average ventilation rate but still yielding high maximum values when occupancy is low. The C-T24 case has the highest rates in the Office based on the minimum requirement. Other than 62tracking, all the ventilation strategies have high per person intake rates at low occupancy relative to the design values. The control approaches based on the Standard 62 rates (C-ZeroMin and C-25%Min) both have average ventilation rates higher than 62tracking. Therefore, while these CO_2 control strategies may have lower ventilation rates during periods of the day, overall they provide more ventilation air than a “perfect” control system, presumably a desirable and conservative outcome from an indoor air quality perspective.

Figure 2 is a plot of the total outdoor air ventilation rate (intake plus infiltration) for a single day in the Office. Note that the two CO_2 control strategies using the Standard 62 rates, C-25%Min and C-ZeroMin, track the idealized case of 62tracking fairly well, with some “underventilation” early in the day and some “overventilation” after occupancy has peaked and towards the end of the day. However, one could argue that one would desire overventilation at these times to “flush out” residual contaminants and that one could tolerate some underventilation early in the day before contaminants have built up. However, overventilation late in the workday can have an energy penalty in hot weather. The use of the terms underventilation and overventilation are only relative to the rates required by the standard or addendum (design values). The C-25%Min case is more conservative, in that it starts the day with higher ventilation rates relative to the 62tracking and 62-ZeroMin cases. The C-62nAreaMin case is even more conservative in the early part of the day, based on its higher minimum ventilation rate, but it does not provide as much ventilation later in the day as the other DCV cases.

	Outdoor Air Intake Rate (neglecting infiltration) L/s•person (cfm/person)				
	Design value	During Occupancy			Min/Max at maximum occupancy
		Average	Minimum	Maximum	
Office					
62/2001	9.4 (20.0)	24.0 (50.9)	9.4 (20.0)	94.0 (200.0)	9.4/9.4 (20.0/20.0)
62tracking	--	9.4 (20.0)	9.4 (20.0)	9.4 (20.0)	9.4/9.4 (20.0/20.0)
C-ZeroMin	--	12.1 (25.7)	0 (0)	65.0 (137.8)	6.1/8.9 (12.9/18.8)
C-25%Min	--	14.7 (31.2)	5.0 (10.5)	70.9 (150.3)	6.6/9.0 (13.9/19.0)
62n	6.8 (14.4)	16.3 (34.6)	6.4 (13.6)	64.1 (135.7)	6.4/6.4 (13.6/13.6)
C-62nAreaMin	--	13.2 (27.9)	5.1 (10.7)	56.9 (120.6)	5.1/6.2 (10.7/13.1)
C-T24	10.8 (23.1)	27.8 (58.8)	10.9 (23.1)	108.9 (230.8)	10.9/10.9 (23.1/23.1)
Conference Room					
62/2001	9.4 (20.0)	49.4 (104.7)	11.8 (25.0)	188.9 (400.1)	11.8/11.8 (25.0/25.0)
62tracking	--	9.4 (20.0)	9.4 (20.0)	9.4 (20.0)	9.4/9.4 (20.0/20.0)
62/Int	--	24.7 (52.4)	5.9 (12.5)	94 (200.0)	5.9/5.9 (12.5/12.5)
C-ZeroMin	--	13.2 (28.0)	0 (0)	26.3 (55.8)	2.0/10.4 (4.2/22.1)
C-25%Min	--	19.7 (41.7)	3.7 (7.9)	53.4 (113.1)	3.7/10.6 (7.9/22.5)
62n	3.9 (8.2)	15.3 (32.5)	3.7 (7.8)	58.5 (124.0)	3.7/3.7 (7.8/7.8)
C-62nAreaMin	--	5.2 (11.0)	1.4 (3.0)	13.4 (28.3)	1.6/3.1 (3.4/6.5)
C-T24	7.1 (15.0)	17.0(36.0)	3.1 (6.6)	42.7 (90.5)	3.2/8.9 (6.8/18.8)
Lecture Hall					
62/2001	7.1 (15.0)	14.7 (31.1)	7.1 (15.0)	35.4 (75.0)	7.1/7.1 (15/15)
62tracking	--	7.1 (15.0)	7.1 (15.0)	7.1 (15.0)	7.1/7.1 (15/15)
62/Int	--	7.4 (15.6)	3.5 (7.5)	17.7 (37.5)	3.5/3.5 (7.5/7.5)
C-ZeroMin	--	10.1 (21.4)	0 (0)	31.6 (67.0)	0.8/7.1 (1.8/15.0)
C-25%Min	--	10.8 (22.9)	2.0 (4.3)	32.1 (68.1)	2.0/7.1 (4.3/15.0)
62n	3.8 (8.0)	7.8 (16.6)	3.8 (8.0)	18.9(40.0)	3.8/3.8 (8.0/8.0)
C-62nAreaMin	--	5.3 (11.2)	0.3 (0.7)	16.5 (35.0)	0.9/3.8 (1.9/8.0)
C-T24	7.1 (15.0)	12.8 (27.1)	0.8 (1.8)	35.4 (75.0)	1.3/7.1 (2.7/15.0)
Classroom					
62/2001	7.1 (15.0)	35.9 (76.1)	7.0 (14.9)	140.3 (297.2)	7.0/7.0 (14.9/14.9)
62tracking		7.0 (14.9)	7.0 (14.9)	7.0 (14.9)	7.0/7.0 (14.9/14.9)
C-ZeroMin		12.1 (25.6)	0 (0)	71.4 (151.2)	0.2/6.9 (0.4/14.7)
C-25%Min		17.1 (36.3)	1.8 (3.8)	84.9 (179.8)	1.8/6.9 (3.8/14.7)
62n	6.7 (14.2)	32.1 (68.1)	6.3 (13.3)	125.6 (266.1)	6.3/6.3 (13.3/13.3)
C-62nAreaMin		15.6 (33.0)	1.6 (3.4)	77.9 (165.0)	1.6/6.2 (3.4/13.1)
C-T24	7.1 (15.0)	20.2 (42.7)	2.2 (4.6)	111.6 (236.5)	2.2/7.0 (4.6/14.9)
Portable Classroom					
62/2001	7.1 (15.0)	21.2 (44.9)	7.0 (14.9)	70.1 (148.6)	7.0/7.0 (14.9/14.9)
62tracking		7.9 (16.7)	7.0 (14.9)	10.9 (23.1)	7.0/7.0 (14.9/14.9)
C-ZeroMin		9.7 (20.6)	0 (0)	43.2 (91.6)	0.5/6.9 (1.0/14.6)
C-25%Min		12.0 (25.4)	1.9 (4.1)	48.4 (102.5)	1.9/6.9 (4.1/14.6)
62n	7.7 (16.2)	21.7 (45.9)	7.2 (15.2)	71.7 (152.1)	7.2/7.2 (15.2/15.2)
C-62nAreaMin		13.3 (28.1)	2.6 (5.6)	51.6 (109.4)	2.6/7.1 (5.6/15.0)
C-T24	8.5 (18.0)	16.0 (33.9)	3.6 (7.6)	60.4 (127.9)	3.6/8.4 (7.6/17.7)
Fast Food					
62/2001	9.4 (20.0)	65.2 (138.1)	9.4 (20.0)	944.0 (2000.0)	9.4/9.4 (20.0/20.0)
62tracking		9.4 (20.0)	9.4 (20.0)	9.4 (20.0)	9.4/9.4 (20.0/20.0)
C-ZeroMin		16.4 (34.8)	0 (0)	87.0 (184.4)	9.2/9.3 (19.5/19.8)
C-25%Min		26.2 (55.5)	9.0 (19.0)	236.1 (500.1)	9.2/9.3 (19.5/19.8)
62n	5.1 (10.8)	35.3 (74.7)	5.1 (10.8)	510.7 (1082.0)	5.1/5.1 (10.8/10.8)
C-62nAreaMin		15.9 (33.6)	4.5 (9.6)	151.8 (321.6)	4.8/5.0 (10.1/10.5)
C-T24	7.1 (15.0)	21.5 (45.5)	7.1 (15.0)	136.2 (288.5)	7.1/7.1 (15.0/15.0)

Table 5 Summary of Ventilation Rates

The Conference Room ventilation results are also presented in Table 5, with the additional case of 62/Int, which implements the intermittent occupancy provision of Standard 62-2001. The highest average intake rate is for 62/2001. Note that the conference room is never occupied at its design value of 50 occupants; the maximum is 40 in these simulations, and this value only occurs for a few hours on Mondays and Wednesdays. Based on the lower and more variable occupancy pattern relative to the Office, the maximum ventilation rates for the CO_2 control approaches are not as high relative to 62-2001 as in the Office. In the Office control cases, the ventilation rates continue to increase over several hours of high occupancy. However, in the Conference Room the occupancy drops before the CO_2 levels get as high, and the ventilation rates are lower on average. Figure 3 presents the Conference Room ventilation rates for Wednesday and Thursday of the simulation period. The most significant difference from the Office results in Figure 2 is seen for the CO_2 control cases that "overshoot" the 62tracking case. This "overshooting" occurs during the short periods of elevated occupancy because these peak occupancy levels are below the design value and the maximum ventilation rate in the CO_2 control algorithm is based on the design occupancy. The Conference Room in fact never achieves the design occupancy, and therefore this overshoot occurs for all the occupancy peaks.

The Lecture Hall results in Table 5 are similar to those for the Conference Room except all the rates are lower given the lower design ventilation rates. Also, the Lecture Hall does attain its design occupancy level, even if only briefly. The Lecture Hall ventilation rates are plotted in Figure 4 and also exhibit the "overshoot" relative to the 62tracking case that is seen in the Conference Room. However, the occupancy peak that occurs after lunch is at the design occupancy level and therefore no overshoot is seen here.

The results for the Classroom and Portable Classroom are similar, with some differences seen due to the lower occupancy density and lower average CO_2 generation rate in the Portable Classroom. These differences generally result in lower per person ventilation rates. The ventilation rates plotted in Figures 5 and 6 for the two spaces exhibit very similar patterns, with the values lower in the Portable Classroom. Note that the C-T24 case has the highest ventilation rates of the control cases in both classrooms, notably so in the Portable Classroom.

The Fast Food Restaurant has an extremely variable occupancy pattern in which the design occupancy pattern is only achieved briefly once during each day. Therefore, the ratio of the average and maximum per person ventilation rates to the design value for the 62/2001 and 62n case are highest for this space. The ventilation rates for Friday and Saturday in this space are plotted in Figure 7.

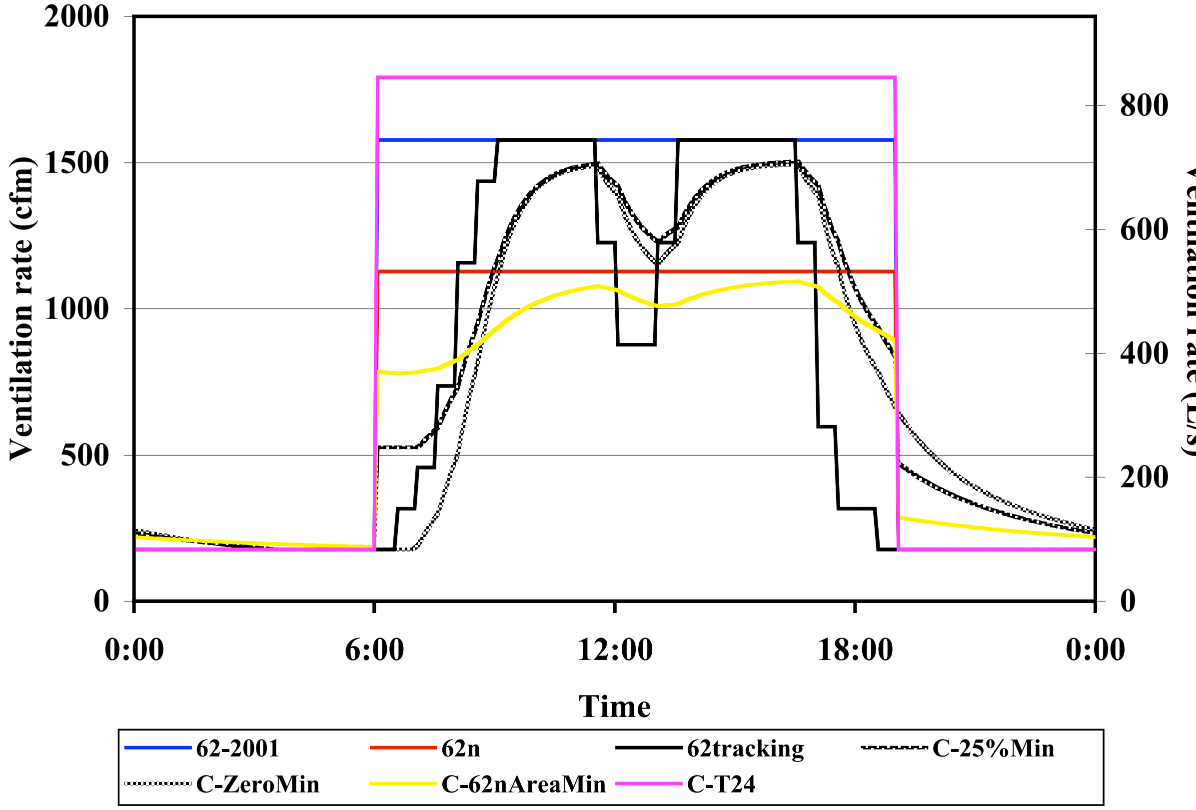


Figure 2 Office Ventilation Rates during Weekday (Friday)

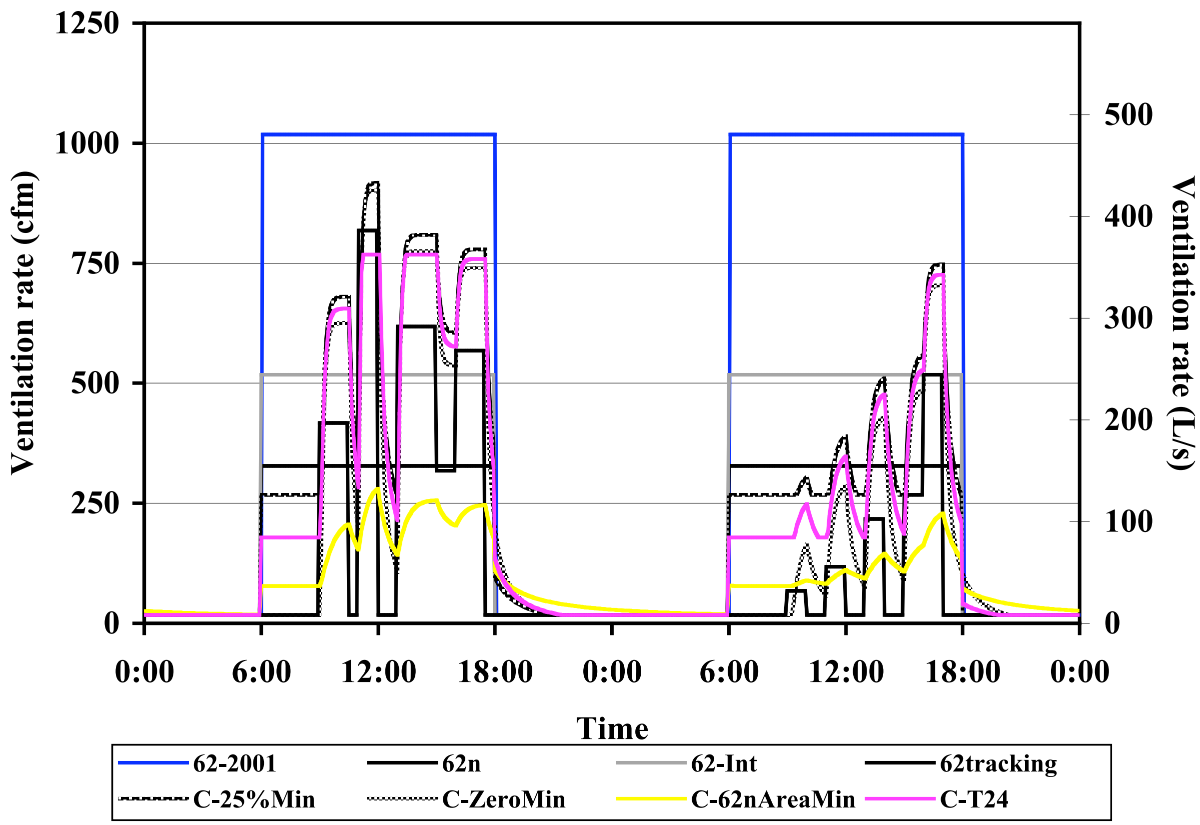


Figure 3 Conference Room Ventilation Rates during Week (Wednesday and Thursday)

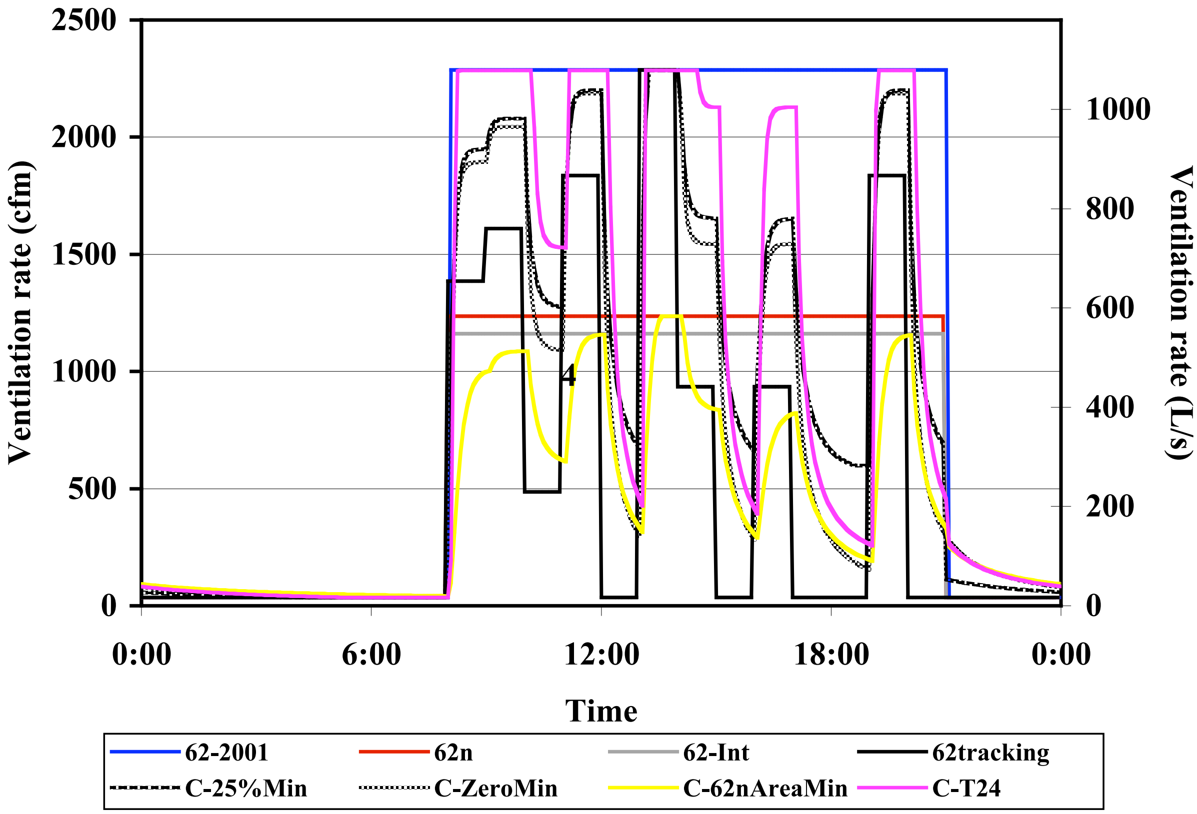


Figure 4 Lecture Hall Ventilation Rates during Week (Friday)

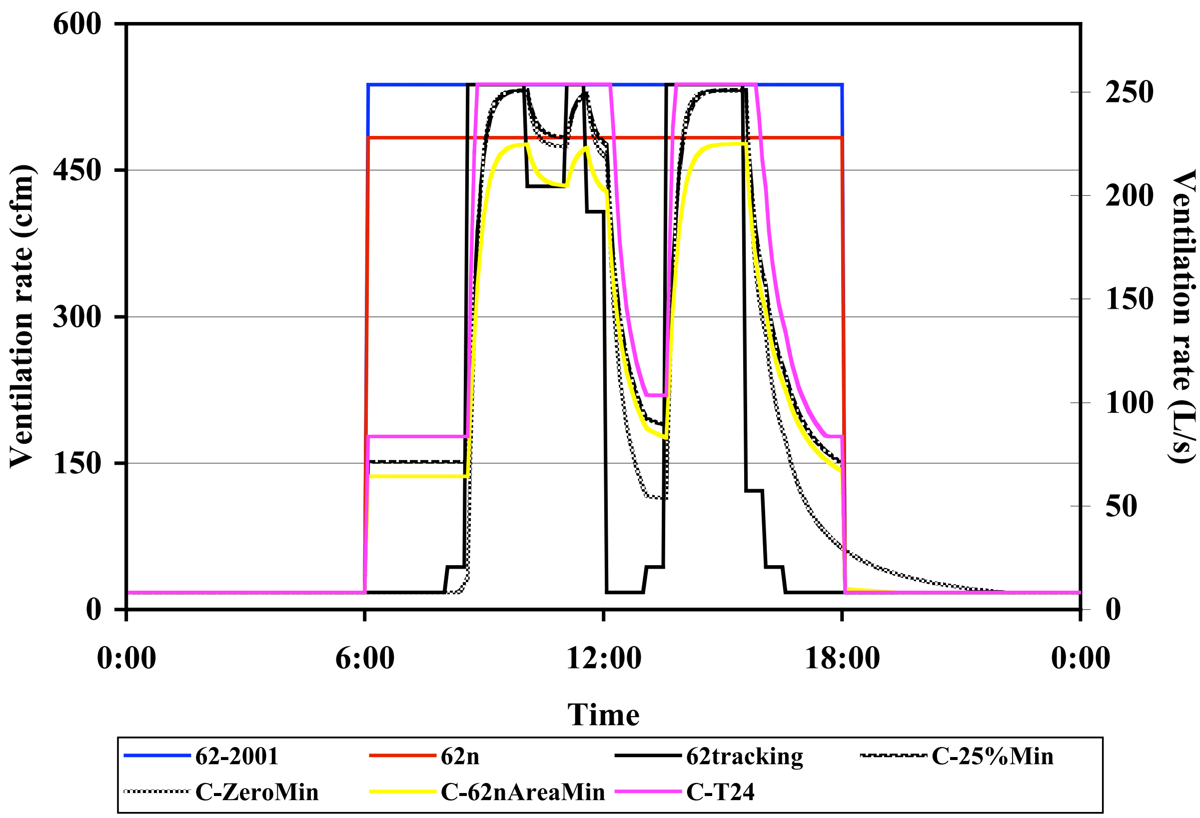


Figure 5 Classroom Ventilation Rates during Week (Friday)

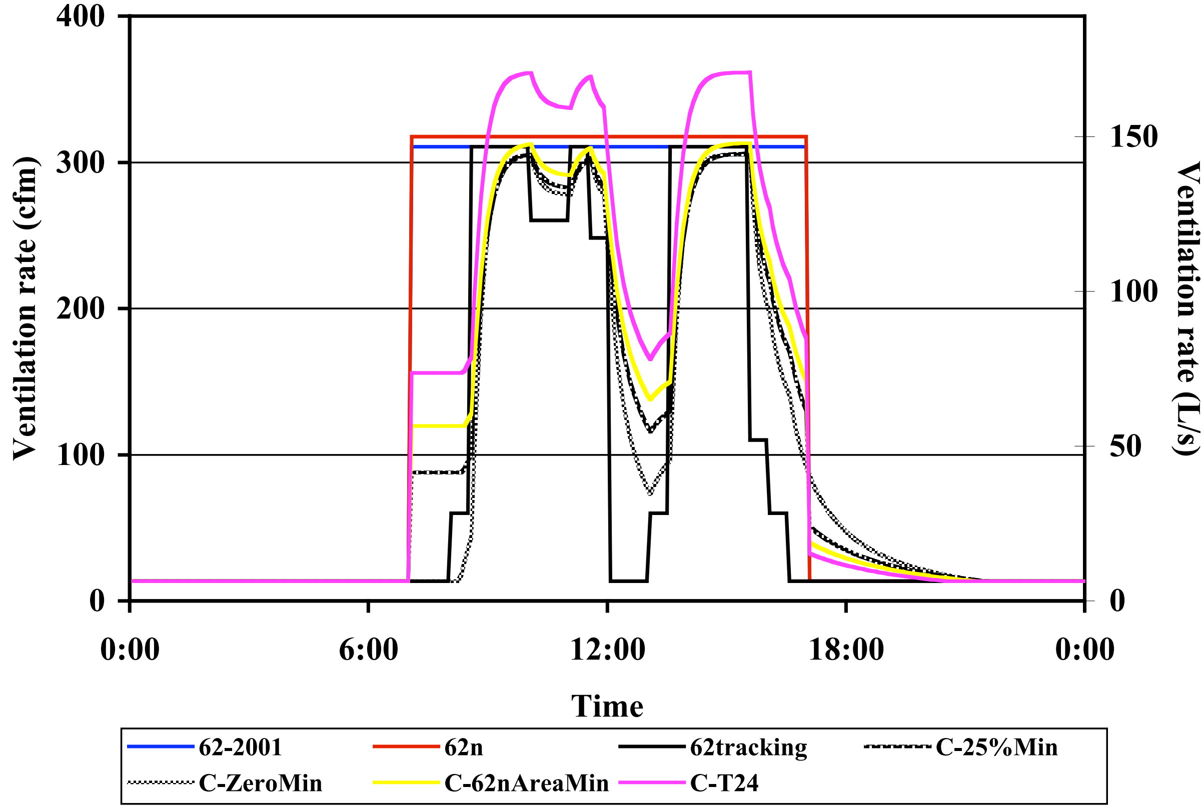


Figure 6 Portable Classroom Ventilation Rates during Week (Friday)

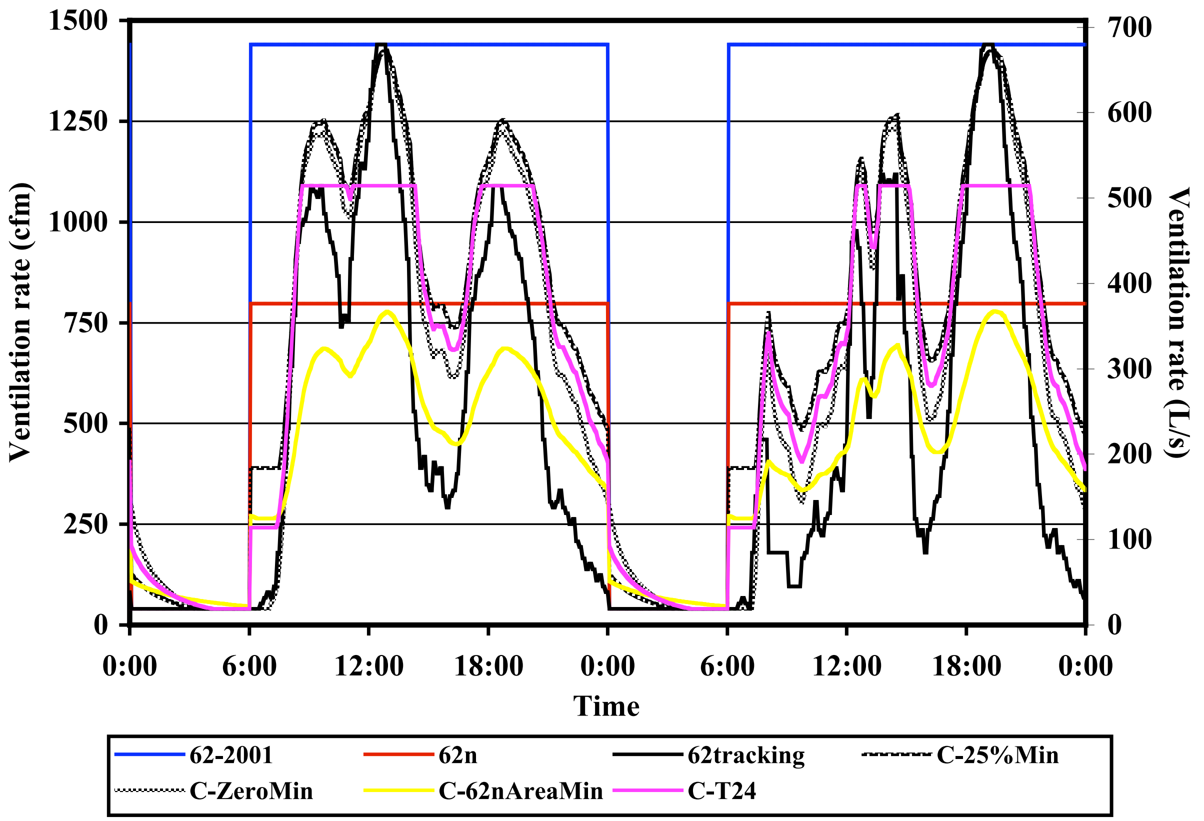


Figure 7 Fast Food Restaurant Ventilation Rates over Friday and Saturday

3.2 Carbon Dioxide Concentration

Table 6 summarizes the indoor CO_2 concentrations for the ventilation strategies and spaces in terms of the average and maximum concentration during occupancy. The first column of the table also presents the steady-state CO_2 concentration for the 62/2001, 62n and Title 24 cases, as well as the 62/Int case when relevant, based on the values in Table 2.

In all spaces, the average and maximum CO_2 concentrations are lower for the 62/2001 cases than for the other cases, except for C-T24 in the Office. This result is expected due to 62/2001 having the highest ventilation rates compared to the 62tracking and the CO_2 control cases. And while 62tracking and the CO_2 control cases have higher average and maximum CO_2 concentrations relative to the 62/2001 case, they are almost always within about 200 mg/m^3 (about 100 parts per million by volume (ppm(v))) of the 62/2001 values. Also, the CO_2 control cases almost always have average and maximum concentrations below the idealized 62tracking case, which indicates good control of occupant-generated contaminants. In all cases the maximum CO_2 concentration during occupancy is less than the steady-state concentration based on the design value in the first column. These differences are generally on the order of 200 mg/m^3 (roughly 100 ppm(v)) except for the Conference Room where the design occupancy is never achieved. The 62/Int case, applicable to only the Conference Room and Lecture Hall, has higher CO_2 concentrations than the other 62-based cases as expected. However, the average concentrations during occupancy are still only about 200 mg/m^3 (roughly 100 ppm(v)) above the 62tracking case.

Figures 8 through 13 present the CO_2 concentrations in each of the spaces over one day (two days in selected cases) for the different ventilation strategies. The CO_2 concentrations in the Office in Figure 3 are fairly similar for the six different ventilation strategies. The 62tracking case is higher during unoccupied periods as the ventilation rate during those periods includes only infiltration, and therefore the post-occupancy CO_2 concentration is elevated relative to the other strategies. This residual concentration builds up during the week, and the biggest differences are seen in this plot for Friday. The two 62n cases, 62n and C-62nAreaMin, have higher concentrations than the other cases during occupancy based on the lower design ventilation rates. The Title 24 DCV case has the lowest CO_2 levels in the Office as expected since it has the highest, and in fact constant, ventilation rates during occupancy. In the other spaces, the 62-2001 case has the lowest CO_2 levels due its having the highest ventilation rates.

There is more variation in CO_2 levels among the ventilation strategies for the Conference Room as seen in Figure 9. The two 62n-based cases, 62n and C-62nAreaMin, have lower per person ventilation rates relative to Standard 62, and therefore the CO_2 concentrations are significantly higher. The 62tracking case again has elevated concentrations during unoccupied periods based on only infiltration occurring during these times. The Lecture Hall in Figure 10 shows the same features as the Conference Room, higher concentrations for the 62n-based cases and elevated concentrations for 62tracking during unoccupied periods. The different ventilation strategies have fairly similar CO_2 concentrations in the two classroom cases (Figures 11 and 12). Finally, the Fast Food Restaurant in Figure 13 also exhibits elevated concentrations for the 62n-based cases and elevated concentrations for 62tracking.

From the concentrations in Table 6 and the figures, one sees that the CO_2 control cases result in CO_2 concentrations that are not very different from those in the cases without CO_2 control. While the CO_2 control cases have higher concentrations, the differences are generally on the order of 200 mg/m^3 (100 ppm(v)).

	Indoor CO_2 concentrations during occupancy			
	Average		**Maximum**	
	mg/m^3	(ppm(v))	mg/m^3	(ppm(v))
Office				
62/2001 (1674 mg/m^3, 930 ppm(v))*	1305	725	1555	864
62tracking	1427	793	1571	873
C-ZeroMin	1413	785	1620	900
C-25%Min	1393	774	1613	896
62n (2061 mg/m^3, 1145 ppm(v))*	1512	840	1854	1030
C-62nAreaMin	1573	874	1908	1060
C-T24 (1546 mg/m^3, 859 ppm(v))*	1231	684	1447	804
Conference Room				
62/2001 (1674 mg/m^3, 930 ppm(v))*	1037	576	1471	817
62tracking	1467	815	1661	923
62/Int (2626 mg/m^3, 1459 ppm(v))*	1291	717	2106	1170
C-ZeroMin	1244	691	1575	875
C-25%Min	1183	657	1561	867
62n (3740 mg/m^3, 2078 ppm(v))*	1553	863	2736	1520
C-62nAreaMin	1962	1090	3240	1800
C-T24 (1991 mg/m^3, 1106 ppm(v))*	1202	668	1694	941
Lecture Hall				
62/2001 (1991 mg/m^3, 1106 ppm(v))*	1436	798	1980	1100
62tracking	1926	1070	1980	1100
62/Int (3262 mg/m^3, 1812 ppm(v))*	2032	1129	3078	1710
C-ZeroMin	1606	892	1980	1100
C-25%Min	1568	871	1980	1100
62n (3123 mg/m^3, 1735 ppm(v))*	1962	1090	2952	1640
C-62nAreaMin	2299	1277	3024	1680
C-T24 (1991 mg/m^3, 1106 ppm(v))*	1469	816	1962	1090
Classroom				
62/2001 (1991 mg/m^3, 1106 ppm(v))*	1559	866	1962	1090
62tracking	1827	1015	1962	1090
C-ZeroMin	1688	938	1980	1100
C-25%Min	1656	920	1980	1100
62n (2113 mg/m^3, 1174 ppm(v))*	1647	915	2088	1160
C-62nAreaMin	1760	978	2124	1180
C-T24 (1991 mg/m^3, 1106 ppm(v))*	1573	874	1944	1080
Portable Classroom				
62/2001 (1532 mg/m^3, 851 ppm(v))*	1262	701	1496	831
62tracking	1418	788	1505	836
C-ZeroMin	1352	751	1519	844
C-25%Min	1332	740	1517	843
62n (1494 mg/m^3, 830 ppm(v))*	1251	695	1480	822
C-62nAreaMin	1310	728	1499	833
C-T24 (1399 mg/m^3, 777 ppm(v))*	1222	679	1384	769
Fast Food Restaurant				
62/2001 (1674 mg/m^3, 930 ppm(v))*	1132	629	1640	911
62tracking	1566	870	1656	920
C-ZeroMin	1314	730	1667	926
C-25%Min	1246	692	1636	909
62n (2461 mg/m^3, 1367 ppm(v))*	1463	813	2322	1290
C-62nAreaMin	1687	937	2412	1340
C-T24 (1991 mg/m^3, 1106 ppm(v))*	1318	732	1908	1060

* Steady-state CO_2 concentration based on the design ventilation rate from Table 2.

Table 6 Summary of Carbon Dioxide Concentrations

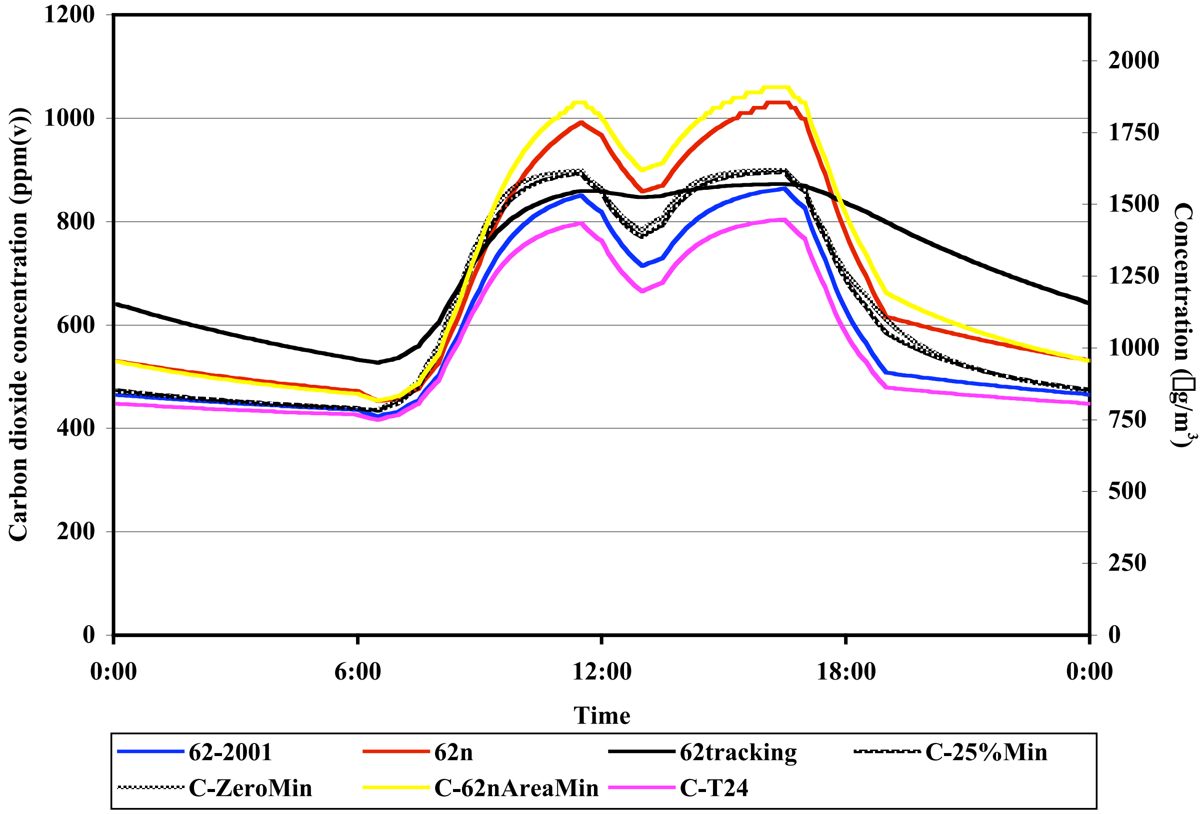


Figure 8 Office CO_2 Concentrations during Weekday (Friday)

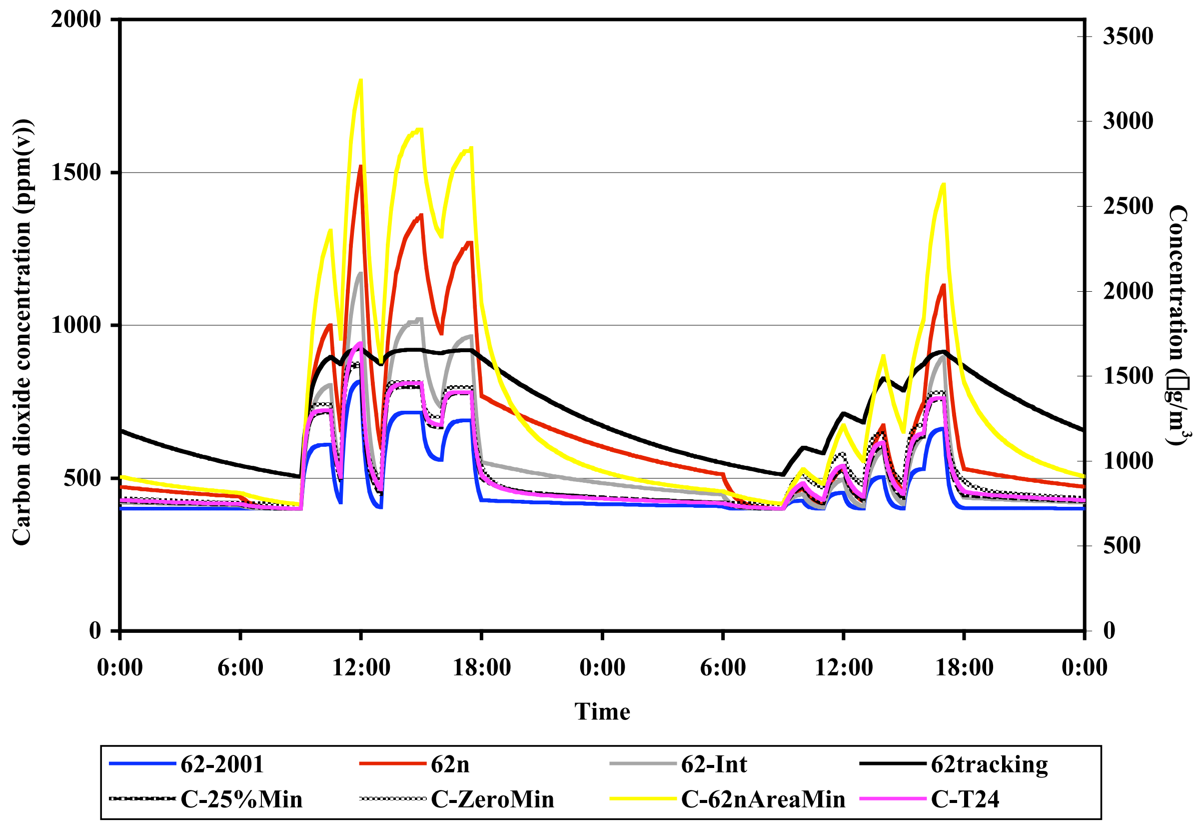


Figure 9 Conference Room CO_2 Concentrations during Week (Wednesday and Thursday)

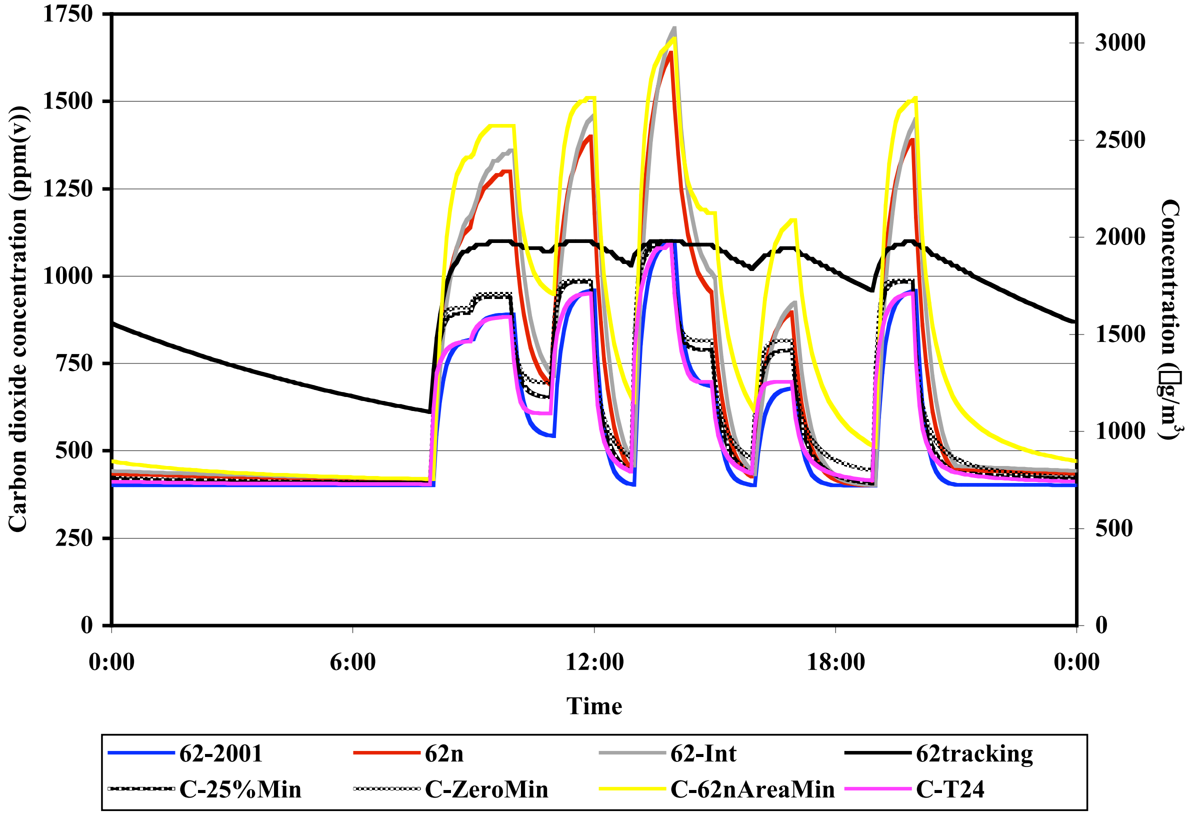


Figure 10 Lecture Hall CO_2 Concentrations during Week (Friday)

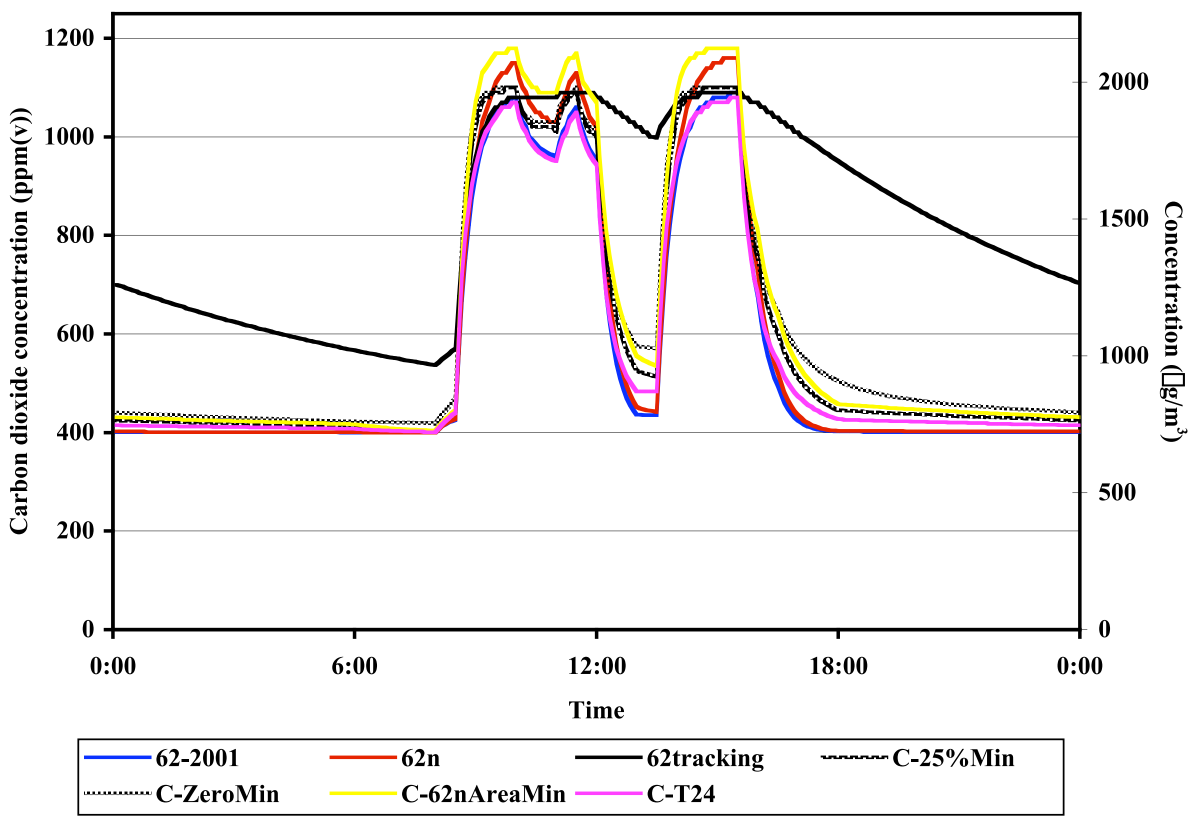


Figure 11 Classroom CO_2 Concentrations during Week (Friday)

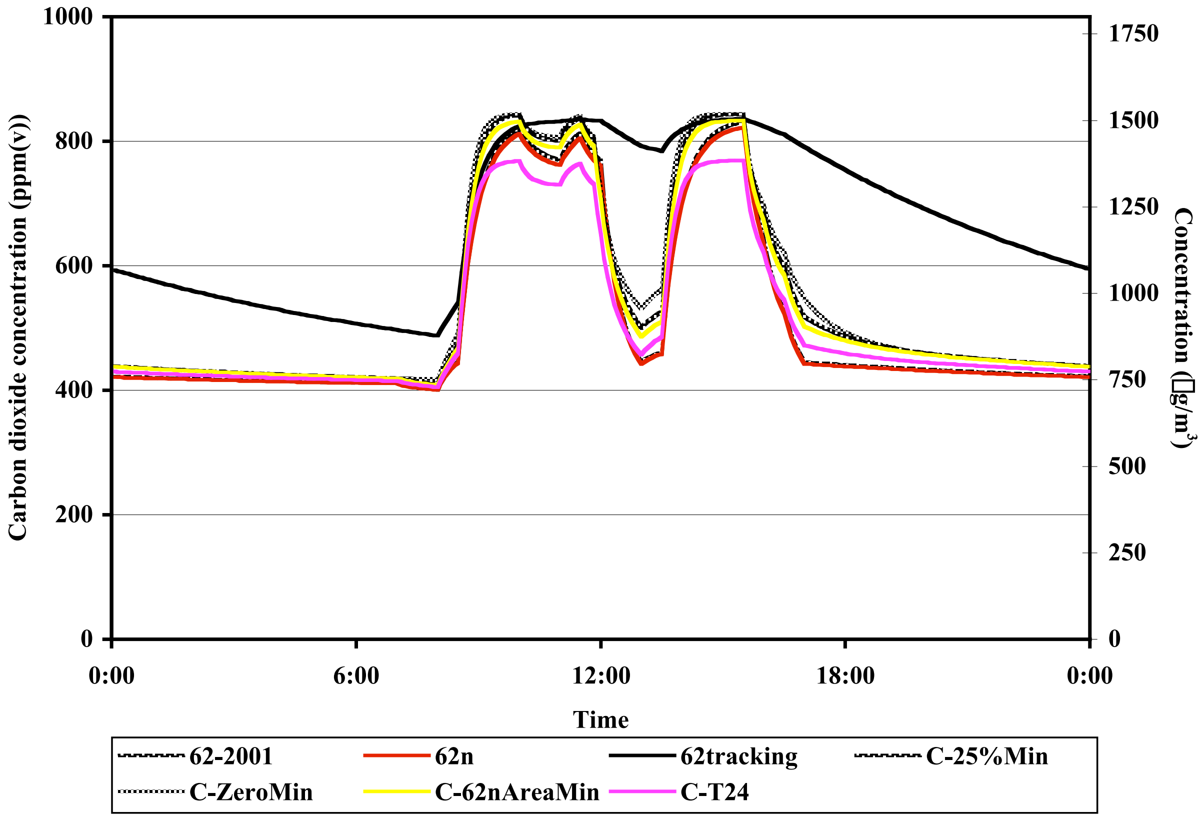


Figure 12 Portable Classroom CO_2 Concentrations during Week (Friday)

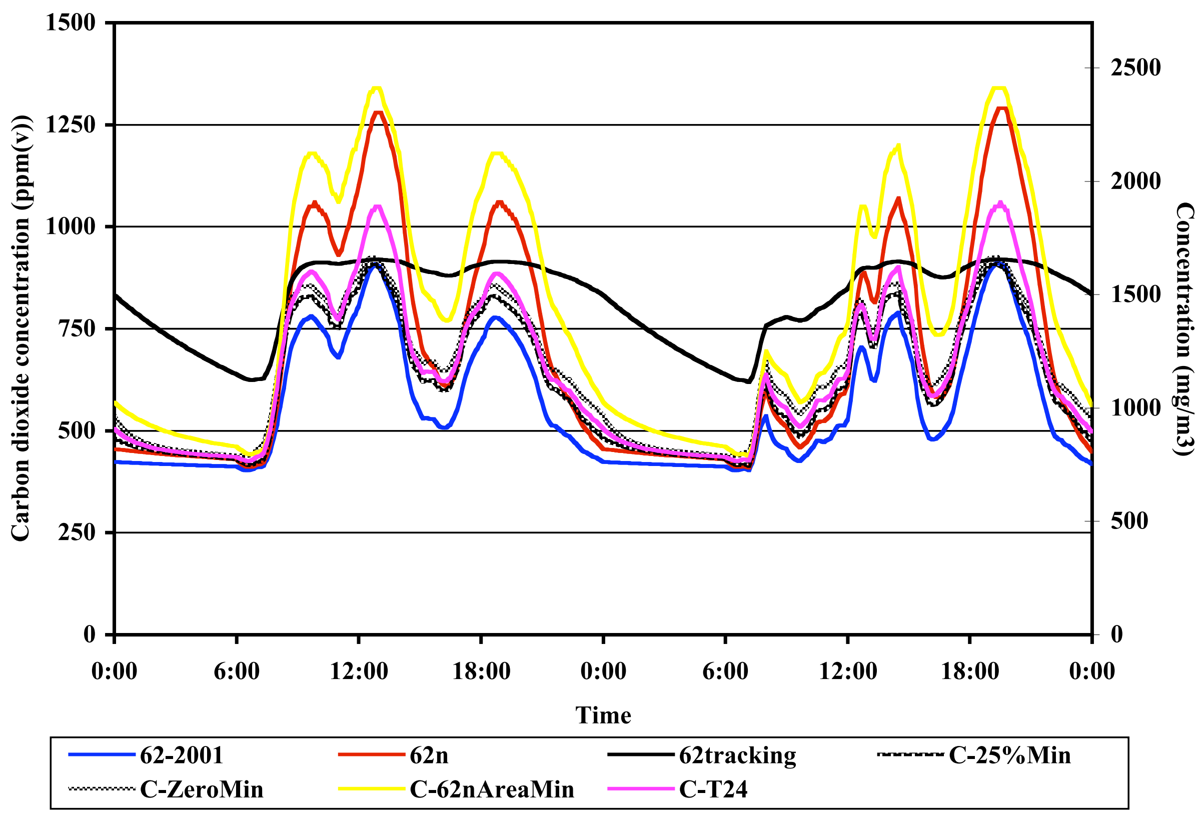


Figure 13 Fast Food Restaurant CO_2 Concentrations over Friday and Saturday

3.3 VOC Concentration

Table 7 summarizes the indoor VOC concentrations for the ventilation strategies in each of the spaces in terms of the average and maximum concentrations during occupancy. The first column of the table also presents the steady-state VOC concentration for the various cases based on the design ventilation rates from Table 2. Figures 14 through 19 present the VOC concentrations in each of the spaces over one day (two in selected cases). Note that the averages in Table 7 are less than 0.4 mg/m^3 in all cases and less than 0.1 mg/m^3 in most cases. While these concentrations are on the low end of those measured in the field, they are dependent on the assumed source strengths during occupied and unoccupied periods and on the assumed infiltration rate during the unoccupied periods. The maximum concentrations are closer to, and in several cases above, 1 mg/m^3 as a result of increases in concentration over unoccupied periods as discussed below.

In all spaces, the average and maximum VOC concentrations are lower for the 62/2001case than for the other cases, except C-T24 in the Office, as expected. And while the 62tracking and CO_2 control cases have higher average concentrations than the 62/2001 case, these averages are generally only two to three times higher and always below 0.4 mg/m^3. Also, the CO_2 control cases almost always have average concentrations that are close to or even below the 62tracking case. If one is willing to assume that 62tracking, which is clearly in compliance with the standard, provides adequate control of building-related contaminants, then the CO_2 DCV cases also control these contaminants on average. The 62n-based CO_2 control case (C-62nAreaMin) also has an average VOC concentration that is generally within a factor of two of the 62n case, again indicating reasonable control of building-related contaminants. The only exception is in the Conference Room where the initial VOC concentration for C-62nAreaMin is elevated at the start of the day. It is worth noting that the average VOC level in non-office spaces with CO_2 control is almost always lower than the VOC level in the Office with the Standard 62-2001 ventilation rate. Therefore, if the VOC level in the Office based on Standard 62 is acceptable, then the level in the other spaces is also acceptable. Of course, this conclusion is based on the same VOC emission rates per unit floor area in all the spaces, which may not always be a good assumption.

The maximum VOC concentrations during occupancy are generally greater than the steady-state VOC concentrations from Table 2 by a factor of two or three, but in some cases by more than an order of magnitude. These maximum concentrations are so much higher due to the increase in VOC concentrations over unoccupied periods when the spaces have an infiltration rate of only 0.1 h^{-1}. Note that the steady-state concentrations are based on the design ventilation rates and are not impacted by the elevated initial concentrations that may exist early in the morning. The maximum concentrations are strongly dependent on the unoccupied infiltration rate and the relative values of the source strength during occupied and unoccupied periods, and therefore the relative values of the different cases are more informative than the absolute values. The highest maximum concentrations are seen for 62tracking or the CO_2 control cases where there is very little ventilation in the early part of occupancy. The maximum concentrations confirm the observation above that if the 62tracking case is considered to control IAQ acceptably then the DCV cases should also be considered acceptable as the maximum concentrations for the DCV cases are near or below the 62tracking cases in all situations.

Figure 14 presents the VOC concentrations in the Office space. The patterns are similar for the different ventilation cases with the exception of the elevated concentrations early in the day for 62tracking and C-ZeroMin, where the VOC source increases as soon as the system comes on but the ventilation rate does not increase until later in the morning. A slight increase in concentration is seen at mid-day for the 62tracking and the CO_2 control cases when the ventilation rate is reduced in response to the lower occupancy. After the system is turned off at 1900, the VOC concentration increases steadily and reaches its maximum value just before the system comes back on the next morning. Figures 15 and 16 shows similar trends for the Conference Room and Lecture Hall, with dramatic increases in concentration at system startup for 62tracking and C-ZeroMin. Once the ventilation rates increase, the concentrations reduce by a factor of 5 to 10. The increases in VOC concentrations during periods of low occupancy are more evident in these spaces than in the Office. The two classrooms show similar results in Figures 17 and 18, as does the Fast Rood Restaurant in Figure 19.

The average and maximum VOC concentrations in Table 7 are both heavily influenced by the elevated concentrations at the start of occupancy. The figures reveal that once the early morning transients die out, the differences between the various ventilation strategies become less significant. Looking at the concentrations in the late afternoon, the ratio of maximum to minimum VOC concentrations for the various cases, neglecting the idealized 62tracking case, is about 1.5 in the Office, at most 3 or 4 in the Conference Room, Lecture Hall and classrooms, and around 2 in the fast food restaurant.

	Indoor VOC concentrations during occupancy (mg/m^3)	
	Average	Maximum
Office		
62/2001 (0.21 mg/m^3)*	0.22	0.58
62tracking	0.34	0.89
C-ZeroMin	0.37	0.96
C-25%Min	0.32	0.80
62n (0.03 mg/m^3)*	0.31	0.67
C-62nAreaMin	0.35	0.74
C-T24 (0.18 mg/m^3)*	0.19	0.55
Conference Room		
62/2001 (0.03 mg/m^3)*	0.03	0.03
62tracking	0.30	0.90
62/Int (0.06 mg/m^3)*	0.06	0.06
C-ZeroMin	0.22	1.06
C-25%Min	0.07	0.12
62n (0.09 mg/m^3)*	0.09	0.09
C-62nAreaMin	0.25	0.50
C-T24 (0.04 mg/m^3)*	0.09	0.20
Lecture Hall		
62/2001 (0.01 mg/m^3)*	0.02	0.28
62tracking	0.04	0.32
62/Int (0.03 mg/m^3)*	0.04	0.34
C-ZeroMin	0.03	0.42
C-25%Min	0.03	0.37
62n (0.03 mg/m^3)*	0.03	0.33
C-62nAreaMin	0.06	0.42
C-T24 (0.01 mg/m^3)*	0.03	0.41
Classroom		
62/2001 (0.06 mg/m^3)*	0.06	0.06
62tracking	0.18	0.99
C-ZeroMin	0.19	1.03
C-25%Min	0.10	0.31
62n (0.06 mg/m^3)*	0.06	0.07
C-62nAreaMin	0.11	0.35
C-T24 (0.06 mg/m^3)*	0.08	0.26
Portable Classroom		
62/2001 (0.09 mg/m^3)*	0.09	0.18
62tracking	0.22	1.04
C-ZeroMin	0.24	1.10
C-25%Min	0.17	0.64
62n (0.08 mg/m^3)*	0.09	0.17
C-62nAreaMin	0.15	0.52
C-T24 (0.07 mg/m^3)*	0.12	0.41
Fast Food		
62/2001 (0.03 mg/m^3)*	0.03	0.06
62tracking	0.10	0.36
C-ZeroMin	0.07	0.31
C-25%Min	0.05	0.18
62n (0.05 mg/m^3)*	0.05	0.11
C-62nAreaMin	0.08	0.20
C-T24 (0.04 mg/m^3)*	0.06	0.21

* Steady-state VOC concentration based on the design ventilation rate from Table 2.

Table 7 Summary of VOC Concentrations

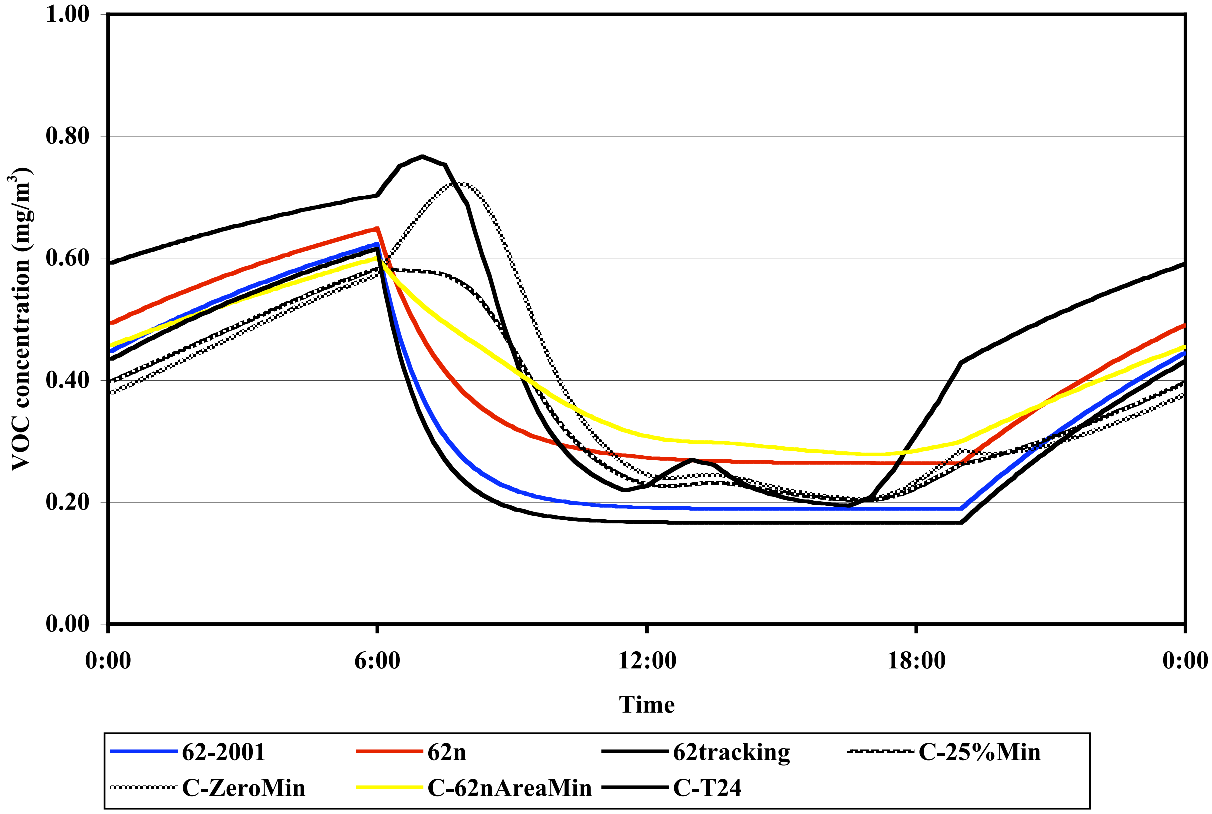


Figure 14 Office VOC Concentrations during Weekday (Friday)

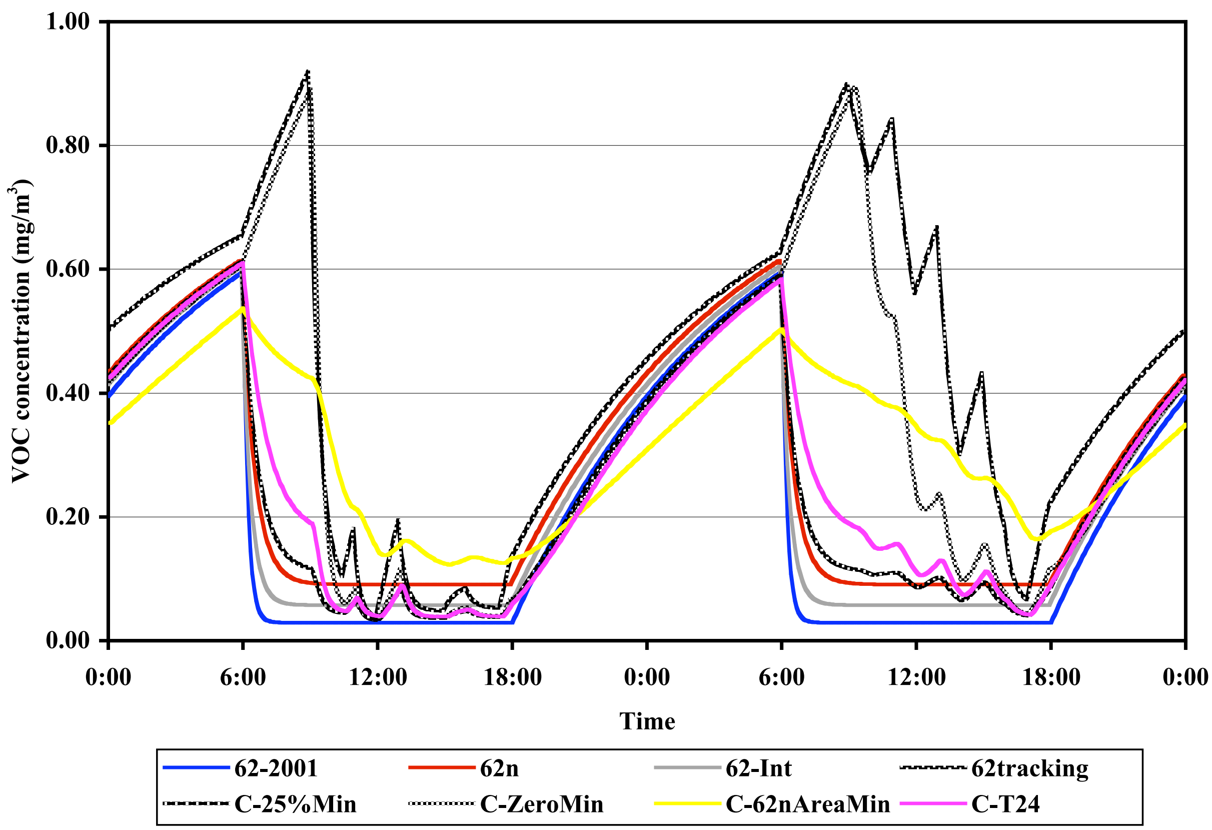


Figure 15 Conference Room VOC Concentrations during Week (Wednesday and Thursday)

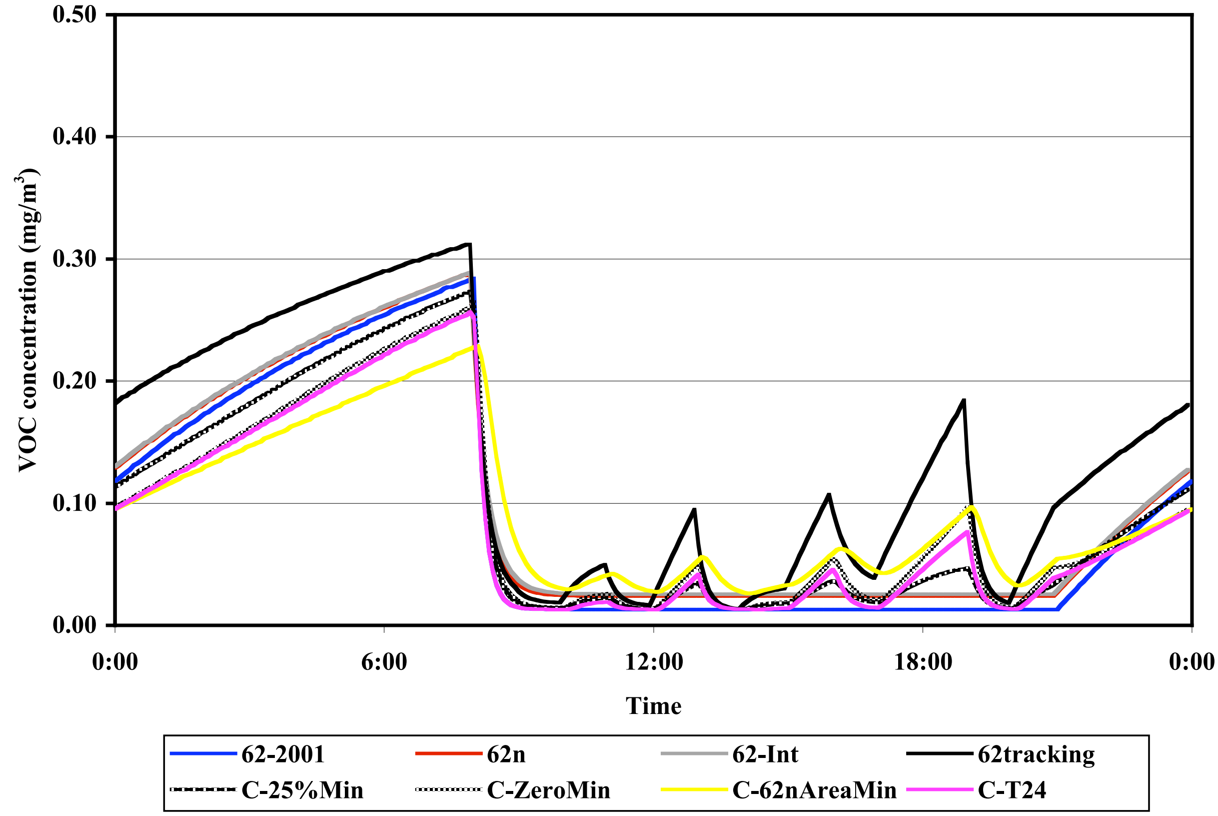


Figure 16 Lecture Hall VOC Concentrations during Week (Friday)

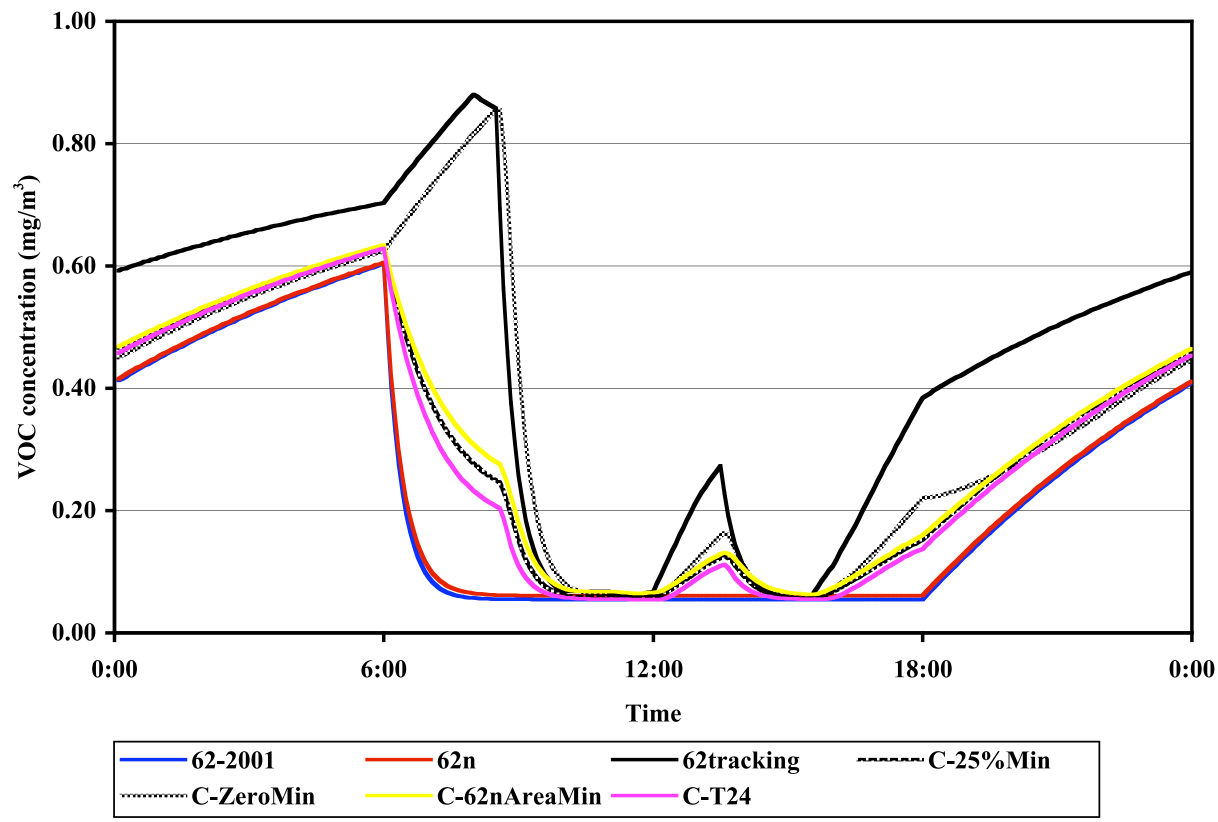


Figure 17 Classroom VOC Concentrations during Week (Friday)

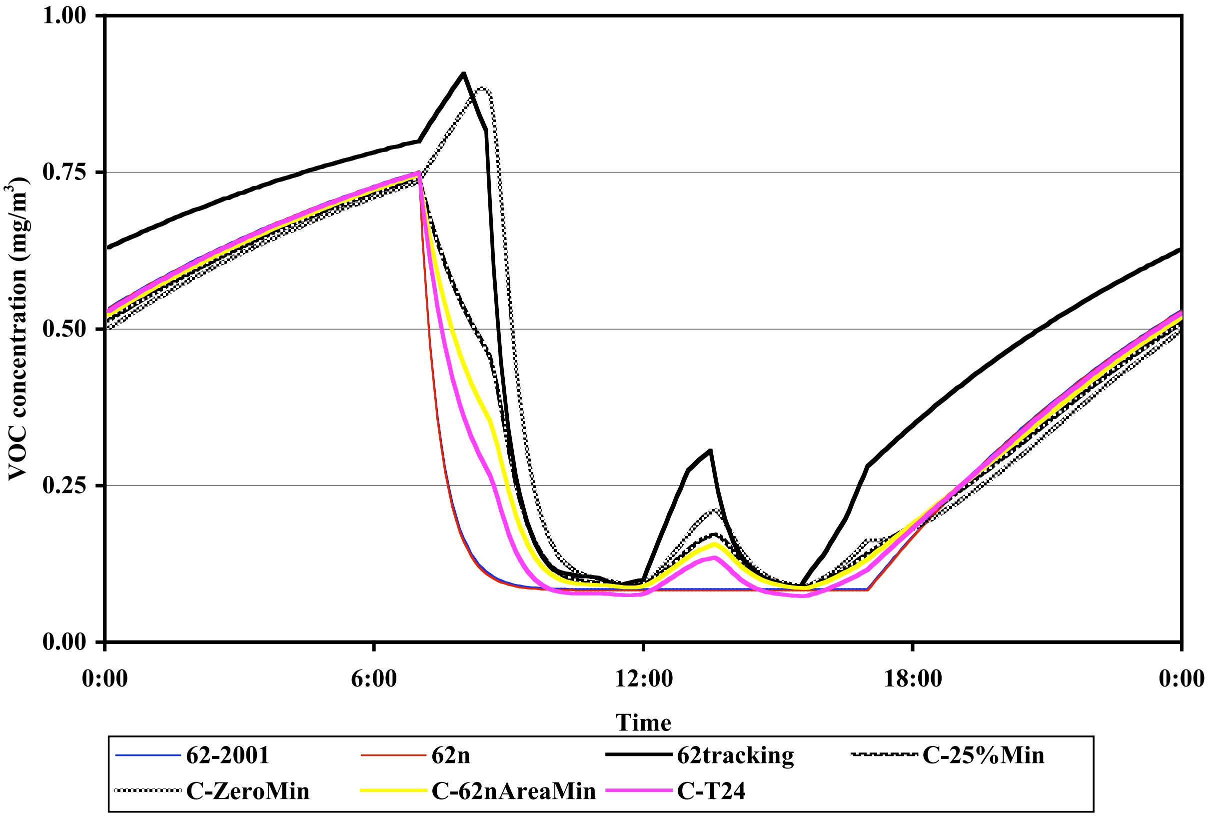


Figure 18 Portable Classroom VOC Concentrations during Week (Friday)

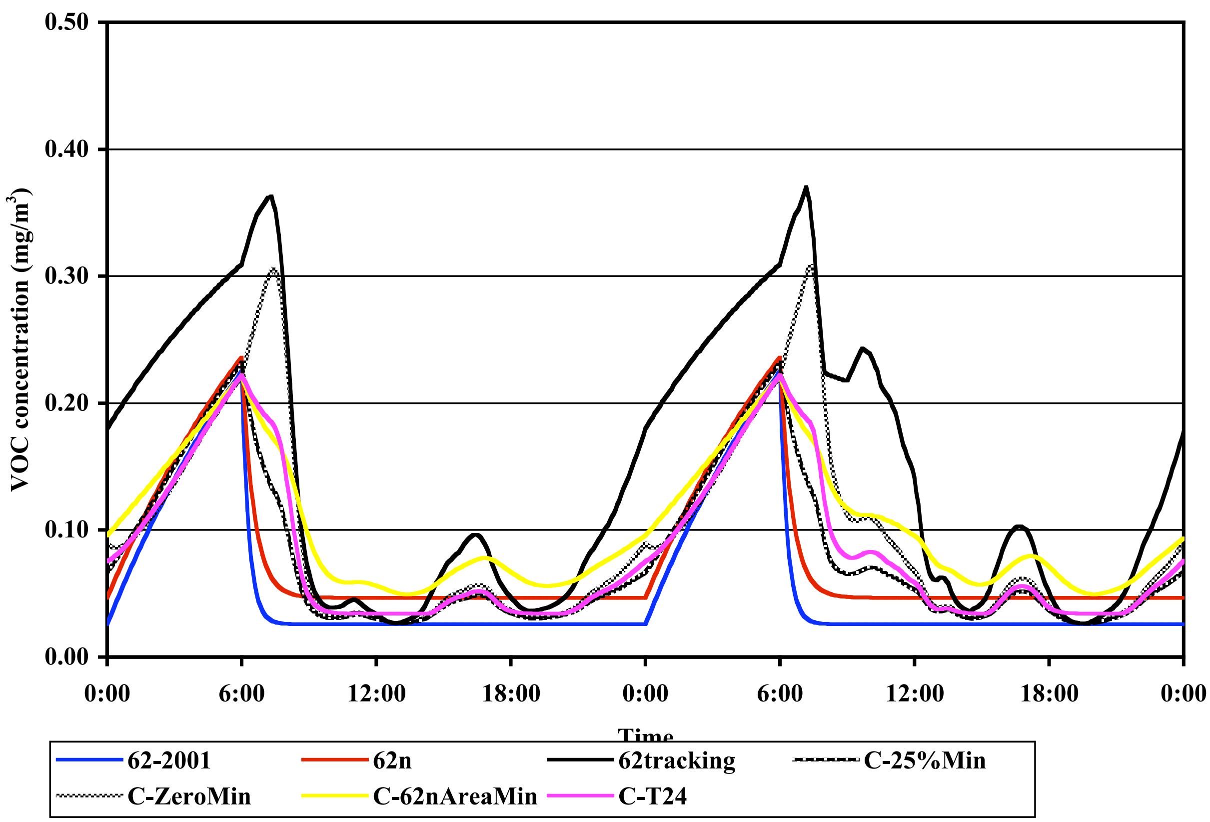


Figure 19 Fast Food Restaurant VOC Concentrations over Friday and Saturday

3.4 Energy Loads

Table 8 summarizes the estimated energy loads associated with ventilation for each of the spaces. For each city, this table presents the annual energy load associated with ventilation for each ventilation strategy in units of MJ/m^2 to account for differences in the sizes of the spaces. Appendix A contains more details on the energy loads, including the heating, sensible cooling and latent cooling for each case. In general, the CO_2 control cases use less energy than the constant ventilation rate cases, and the 62n case uses less than 62/2001 except in the Portable Classroom where the 62n ventilation rate is higher. The magnitude of these reductions in a particular city and space combination is a fairly complex function of climate (relative amounts of heating and cooling), ventilation rate per unit floor area as shown in Table 2 (which also impacts the heating load via the balance point), internal heat loads shown in Table 3 (which also impacts balance point) and occupancy patterns.

In the Office space, 62tracking and the two 62-based CO_2 control cases have energy loads that are roughly 20 % lower than the straight 62-2001 case. (The loads in San Francisco are so low in the Office space that the reductions are not discussed.) Compared to 62/2001, the addendum 62n rates decrease the ventilation-induced energy load by 30 % to 60 % depending on the city, with the largest reduction in the heating-dominated Minneapolis climate. Implementing CO_2 control under 62n leads to further variable reductions in the energy loads among the different cities, ranging from around 10 % to 30 %. For the Office space, the biggest reductions in energy under CO_2 control occur in Minneapolis, which is a heating dominated climate. These reductions are larger under heating due to the combined impact of lower ventilation rates under CO_2 control and decreased balance point temperatures due to these lower rates. Another reason for larger reductions under heating is that the ventilation rates tend to be lower early in the day when the outdoor temperature is lower. The energy load increases in the Office under C-T24, but as noted earlier that is not really a DCV case in the Office and in fact has a higher ventilation rate than required by Standard 62-2001.

The reductions in energy load for the CO_2 control cases, including C-T24, are much larger in the Conference Room and Lecture Hall given the higher design ventilation rates per unit floor area and impact of these rates on balance point temperature. The reduction in energy load from the 62/2001 to 62n cases is also much larger than in the Office, more than 50 % in all cases and as high as 80 %. Implementing CO_2 control under 62n leads to further reductions in energy load. The results for the classrooms are similar in relative magnitude to the changes seen in the other cases, with the exception of the difference between 62n and 62/2001. As noted earlier, the ventilation rate under 62n in the Portable Classroom is actually higher than under 62/2001, leading to a slight increase in the energy load. Otherwise, implementing CO_2 control under Standard 62, Title 24 or 62n results in energy reductions from around 30 % to 50 % depending on the city. Implementing CO_2 control under Standard 62 or Title 24 in the Fast Food Restaurant reduces the energy load by around 40 % to as high as 75 %, again depending on the city. The lower rates under addendum 62n reduce the energy load by around 50 % relative to 62/2001, with an 86 % reduction in Los Angeles due to a greater relative reduction in the heating load compared to the other cities.

The energy reductions for C-62nAreaMin relative to 62/2001 are as high as 80 % to 90 % in many cases based on the lower rates in addendum 62n. Also, the percent reductions in the spaces studies are much greater in the mild climates than the more extreme climates.

	Annual Energy Load due to Ventilation (MJ/m^2)					
	Bakersfield	Los Angeles	Sacramento	San Francisco	Miami	Minneapolis
Office						
62/2001	30	6	18	1	117	63
62tracking	24	5	15	1	85	34
C-ZeroMin	26	5	17	1	87	35
C-25%Min	27	5	17	1	93	37
62n	20	4	12	1	79	18
C-62nAreaMin	19	4	12	1	71	13
C-Title24	35	7	21	1	135	94
Conference Room						
62/2001	357	173	348	298	670	727
62tracking	71	25	66	44	147	148
62/Int	169	63	162	127	332	356
C-ZeroMin	111	41	104	73	228	233
C-25%Min	151	56	145	106	303	320
62n	93	22	89	53	205	213
C-62nAreaMin	30	4	22	4	87	71
C-Title24	129	48	122	85	248	280
Lecture Hall						
62/2001	1049	528	1010	931	1943	2168
62tracking	383	142	362	292	790	841
62/Int	464	127	437	322	962	1041
C-ZeroMin	568	219	537	428	1143	1231
C-25%Min	645	248	614	502	1278	1395
62n	508	157	479	372	1025	1117
C-62nAreaMin	242	40	215	95	620	618
C-Title24	714	302	687	585	140	1521
Classroom						
62/2001	197	56	194	132	406	446
62tracking	81	16	74	34	203	202
C-ZeroMin	97	19	87	43	236	228
C-25%Min	105	21	93	45	271	264
62n	168	43	166	105	364	397
C-62nAreaMin	88	16	80	33	242	230
C-Title24	119	23	108	53	300	303
Portable Classroom						
62/2001	108	36	106	79	219	236
62tracking	62	17	57	35	135	138
C-ZeroMin	64	17	61	37	143	146
C-25%Min	71	17	65	40	157	163
62n	110	38	109	81	222	240
C-62nAreaMin	76	17	72	44	166	175
C-Title24	94	27	92	63	199	212
Fast Food						
62/2001	1021	514	1041	1018	1875	2171
62tracking	362	94	326	229	833	847
C-ZeroMin	516	158	489	381	1109	1204
C-25%Min	574	180	550	442	1228	1357
62n	435	74	421	254	995	1106
C-62nAreaMin	222	29	174	50	679	672
C-Title24	490	125	465	345	1090	1178

Table 8 Summary of Energy Loads

4. DISCUSSION

The objective of this study was to examine the ventilation, indoor air quality and energy impacts of CO_2 demand controlled ventilation in a number of different space types and climates based on the project goal of developing application guidance for potential users of CO_2 DCV. The results indicate that these impacts are dependent on the details of the spaces including occupancy patterns, ventilation rate requirements and ventilation system operating schedule as well as the assumptions used in the analysis, including contaminant source strengths and system-off infiltration rates. The results and conclusions presented in this report are therefore specific to the cases studied; however, some general conclusions can be reasonably made and the methodology can be extended to other cases and even used in the design process as discussed below.

In terms of the ventilation rates, the simulations results yield the expected result that basing design ventilation rates on design occupancy levels results in "overventilation" for potentially many hours depending on the occupancy schedule. While ventilating at the design rate, even under low occupancy, may have indoor air quality benefits in terms of better dilution of indoor contaminant sources, there is an energy penalty. The CO_2 control cases help avoid such periods of overventilation, but generally result in relatively low ventilation rates early in the day when occupancy is low. These low rates result in contaminant buildup, particularly of those contaminants associated with the building, including potential exposure to contaminants that may have built up overnight when the system was off. The extent of such contaminant buildup is highly dependent on the source strengths in the unoccupied building, for which only very limited data is available, and fan off infiltration rates, which are highly building specific and weather dependent. Therefore it is very hard to generalize about early occupancy exposure, other than stating that CO_2 control strategies with a nonzero base ventilation rate, such as C-25%Min, C-62nAreaMin and C-T24, will help to temper such exposure. Pre-occupancy "flush out" strategies (a requirement of Title 24, but not Standard 62-2001) may also be helpful in lessening such exposure, but need to be considered for the given space and climate, as early morning ventilation can have energy implications that depend on temperature and humidity variations over the day.

As expected, the 62n rates are significantly lower than the 62/2001 rates for all but the two classroom spaces. While this reduction has generated some questions based on potential IAQ concerns, the CO_2 and VOC simulations provide some insight into this question.

While CO_2 is not a contaminant of concern at typical indoor levels, it has become viewed as an indicator of occupant-generated contaminants and is useful in this respect if the limitations are understood. In particular, it provides information on the acceptability of the space in terms of odor from human bioeffluents and perhaps the level of other occupant generated contaminants, but is not a comprehensive indicator of overall indoor air quality. Many have come to view an indoor CO_2 concentration of 1800 mg/m^3 (1000 ppm(v)) as a threshold separating good and bad indoor air quality, but in reality 1800 mg/m^3 (1000 ppm(v)) CO_2 has no significance from a health or comfort perspective (ASTM 2002) and is only of interest based on it being the expected steady-state concentration at ventilation rates of about 8 L/s (17 cfm) per person and an outdoor concentration of 540 mg/m^3 (300 ppm(v)). Nonetheless, the average and maximum indoor CO_2 concentrations serve as metrics for comparing the different cases.

As expected, ventilating at the Standard 62-2001 rate whenever the system is operating results in the lowest CO_2 concentrations, except for the offices where Title 24 results in lower concentrations. However, the CO_2 control cases increase the average and maximum CO_2 concentrations during occupancy by only about 180 mg/m^3 (100 ppm(v)). In terms of the impact on bioeffluent perception, 180 mg/m^3 (100 ppm(v)) is not very significant. Specifically, based on the relationship between percent of occupants dissatisfied with human bioeffluents and CO_2 concentration (ECA 1992), a change of 180 mg/m^3 (100 ppm(v)) corresponds to an increase in the percent dissatisfied of only about 2 %.

The use of ventilation rates based on addendum 62n resulted in more significant increases in indoor CO_2 levels. The average concentrations increased from 180 mg/m^3 (100 ppm(v)) or less to about 540 mg/m^3 (300 ppm(v)) in the Conference Room. The increases in the maximum concentration were larger, up to 900 mg/m^3 (500 ppm(v)) in the Lecture Hall and 1260 mg/m^3 (700 ppm(v)) in the Conference Room. These increases in the maximum concentrations are more significant but still below any level of concern based on health (ASTM 2002). None of the maximum concentrations exceeded 3060 mg/m^3 (1700 ppm(v)), which corresponds to about 35 % dissatisfaction with odor from human bioeffluents on the part of unadapted visitors to a space and about 12 % dissatisfaction on the part of adapted occupants (ECA 1992). The term adapted refers to the fact that people become accustomed to body odors relatively quickly, in less than a minute, and therefore express lower levels of dissatisfaction at the same level of body odor (or CO_2) than individuals who have not yet become adapted to these odors.

For some of the CO_2 control scenarios, the indoor CO_2 level built-up during the week, but this was due to the value assumed for infiltration when the system was off. Note that while there is not a great deal of data on commercial building infiltration rates, the value used in these simulations is likely conservatively low for most building-climate combinations. In fact, a low value was selected intentionally to highlight the impact of contaminant build-up during unoccupied periods. The magnitude of the build-up depended on the details of the control algorithm, but was rarely more than about 180 mg/m^3 (100 ppm(v)).

Indoor VOC concentrations were calculated to assess the impact of CO_2 control on non-occupant generated contaminants, for example those emitted by building materials and furnishings. Based on the assumed emission rates, which were not particularly low relative to the limited data from field studies, the average indoor VOC levels during occupancy were always less than 0.4 mg/m^3 and less than 0.1 mg/m^3 in most cases. While the lack of definitiveness of the VOC emission rate values limits the reliability of the predicted VOC levels in absolute terms, the relative comparison between cases is far more reliable. Using the 62tracking case, which complies with Standard 62-2001, as a baseline, all the other cases have average VOC concentrations that are close to or below this idealized case. If one is willing to accept that 62tracking provides adequate control of building-related contaminants, then the CO_2 DCV cases also control these contaminants on average. The CO_2 control cases had higher VOC concentrations than the reference cases based on Standard 62-2001 and addendum 62n, with the greatest increase in the Conference Room based on its low occupancy early in the day. The average concentrations, and more so the maximum concentrations, were heavily influenced by the build-up in concentration during unoccupied hours, which in turn depend on the values, assumed for the fan-off infiltration rate and VOC emission rate. As discussed earlier, these elevated concentrations early in the day can be tempered by a nonzero minimum ventilation rate under CO_2 control (as with C-25%Min, C-62nAreaMin and C-Title24) or with an early morning flush-out. That latter strategy was not evaluated as part of this project, but this simulation approach could be used to investigate its potential benefits. Note that while the VOC results are dependent on the assumed emission rates, they can be scaled up or down linearly for other emission rates as long as the two-to-one ratio of occupied to unoccupied emission rate is maintained.

The annual energy load reductions due to the use of CO_2 control were significant in most of the cases, ranging from 10 % to 80 % depending on the space type, climate and ventilation strategy. For the Office , the reductions are generally around 20 %, given the relatively stable occupancy pattern in that space relative to some of the others. Spaces with more variability in occupancy, such as the Conference Room and Lecture Hall, exhibit larger energy reductions. The energy load reductions associated with the use of addendum 62n relative to the ventilation requirements in Standard 62-2001 are as large as 30 % to 50 % in the spaces where the 62n rates are lower. However, in the two classroom spaces, the 62n rates are similar to those based on Standard 62-2001 and therefore no significant difference is seen. The most significant reductions are seen for the 62n DCV case relative to the Standard 62 baseline case.

Taking a closer look at the California climates, these results indicate that CO_2 DCV is not likely to provide much energy benefit in offices in the milder climates (Los Angeles and San Francisco) for the relatively stable occupancy patterns used in this study. However in the more "severe" climates of Bakersfield and Sacramento, the savings in the Office were more significant. The spaces with more variable occupancy (Conference Room, Lecture Hall and Fast Food Restaurant) resulted in significant energy savings in all the climates studied. The energy savings in the classroom spaces are strongly dependent on the system operating schedule versus the occupancy schedule, and while significant load reductions were seen in this study, application of CO_2 DCV in classrooms may require more careful consideration.

This study has employed an approach to assessing ventilation, IAQ and energy impacts of different ventilation strategies using the control simulation capabilities of the CONTAMW program. As noted above, this methodology can be applied to other spaces, climates and ventilation strategies to investigate a number of other issues of interest. In particular, the impacts of different VOC source strengths in different spaces and variable emissions patterns over the day would be of interest. Also, the impacts of occupancy levels both lower and greater than those assumed in the design, which does occur in practice, would be worth considering.

5. ACKNOWLEDGEMENTS

This report was supported by the California Energy Commission through the Architectural Energy Corporation. The authors acknowledge the contributions of Vern Smith of the Architectural Energy Corporation and Steven Taylor of Taylor Engineering.

6. REFERENCES

Anderson, K, JV Bakke, O Bjørseth, C-G Bornehag, G Clausen, JK Hongslo, M Kjellman, S Kjærgaard, F Levy, L Mølhave, S Skerfving and J Sundell. 1997. TVOC and Health in Non-Industrial Indoor Environments. Report from a Nordic Scientific Consensus Meeting at Långholmen in Stockholm, 1996. Indoor Air, Vol 7:78-91.

ANSI/ASHRAE/IESNA. 1999. ANSI/ASHRAE/IESNA Standard 90.1-1999: Energy Standard for Buildings Except Low-Rise Residential Buildings. American Society of Heating, Refrigerating, and Air Conditioning Engineers and Illuminating Engineering Society of North America.

ASHRAE. 1997. WYEC2: Weather Year for Energy Calculations 2. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASHRAE. 2001a. ANSI/ASHRAE Standard 62-2001, Ventilation for Acceptable Indoor Air Quality. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASHRAE 2001b. ASHRAE Handbook of Fundamentals. American Society of Heating, Refrigerating, and Air Conditioning Engineers.

ASTM. 2002. D6245-98 (2002) Standard Guide for Using Indoor Carbon Dioxide Concentrations to Evaluate Indoor Air Quality and Ventilation. American Society for Testing and Materials.

Brandemuehl, MJ and JE Braun. 1999. The Impact of Demand-Controlled and Economizer Ventilation Strategies on Energy Use in Buildings. ASHRAE Transactions 105 (2): 39-50.

Brown, S, MR Sim, MJ Abramson and CN Gray. 1994. Concentrations of Volatile Organic Compounds in Indoor Air - A Review, p. 123-134. Indoor Air, Vol. 4.

Carpenter, SC. 1996. Energy and IAQ Impacts of CO2-Based Demand Controlled Ventilation. ASHRAE Transactions, Vol. 102 (2): 1183-1197.

CEC. 2001. Energy Efficiency Standards for Residential and Nonresidential Buildings. California Energy Commission, P400-01-024.

Daisey, JM, AT Hodgson, WJ Fisk, MJ Mendell, and J Ten Brinks. 1994. Volatile Organic Compounds in Twelve California Office Buildings: Classes, Concentrations, and Sources, p. 3557-3562. In Atmospheric Environment. Vol. 28, No. 22.

Dols, WS and GN Walton. 2002. CONTAMW 2.0 User Manual. National Institute of Standards and Technology, NISTIR 6921.

ECA (European Collaborative Action). 1992. Guidelines for Ventilation Requirements in Buildings, Report No. 11, EUR1449EN. Luxembourg Office for Official Publications of the European Communities.

Emmerich, SJ, JW Mitchell and WA Beckman. 1994. Demand-Controlled Ventilation in a Multi-Zone Office Building. Indoor Environment, Vol. 3: 331-340.

Emmerich, SJ and AK Persily. 2001. State-of-the-Art Review of CO_2 Demand Controlled Ventilation Technology and Application. National Institute of Standards and Technology, NISTIR 6729.

Enermodal. 1995. An Evaluation of the Effect of CO2-Based Demand-Controlled Ventilation Strategies on Energy Use and Occupant-Source Contaminant Concentrations. ASHRAE Research Project 740-TRP. Enermodal Engineering Limited.

EPA. 1999. Exposure Factors Handbook. U.S. Environmental Protection Agency, EPA/600/C-99/001.

Hadwen, GE, JF McCarthy, SE Womble, JR Girman and HS Brightman. 1997. Volatile Organic Compound Concentrations in 41 Office Buildings in the Continental United States. In Proceedings of Healthy Buildings/IAQ'97. Volume 2: 465-470.

Knapp, CL, TL Stoffel and SD Whitaker. 1980. Insolation Data Manual. Solar Energy Research Institute, SERISP-755-789.

Levin, H. 1995. Emissions Testing Data and Indoor Air Quality. Second International Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings, 1: 465-482.

Marion W and K Urban. 1995. User's Manual for TMY2s, Typical Meteorological Years, Derived from the 1961-1990 National Solar Radiation Data Base. National Renewable Energy Laboratory, NREL/SP-463-7668, E95004064.

Persily, AK. 2001. Addendum 62n: Revising the Ventilation Rate Procedure. ASHRAE Journal 43 (9): 12-13.

Schell, MB, SC Turner and RO Shim. 1998. Application of CO_2-Based Demand-Controlled Ventilation Using ASHRAE Standard 62: Optimizing Energy Use and Ventilation. ASHRAE Transactions 104 (2): 1213-1225.

Schell, MB and D Int-Hout. 2001. Demand Control Ventilation Using CO_2. ASHRAE Journal 43 (2): 18-29.

Sowa J. 2002. CO_2-Based Occupancy Detection for On-Line Demand Controlled Ventilation Systems. Proceedings of Indoor Air 2002, Vol. 3: 334-339.

Wolkoff, P. 1995. Volatile Organic Compounds – Sources, Measurements, Emissions, and The Impact on Indoor Air Quality. Indoor Air. Supplement No. 3/95.

Appendix A: Details of Energy Simulation Results

This appendix presents the energy simulation results for the six space types for each ventilation control strategy in each city. The tables that follow break down the total energy presented in the body of the report into heating energy and sensible and latent cooling. A separate table is included for each of the six cities.

	Heating (MJ)	Sensible cooling (MJ)	Latent cooling (MJ)	Total cooling (MJ)	Total heating & cooling (MJ)	Total heating & cooling (MJ/m^2)
Office						
62/2001	0	29,800	200	30,000	30,000	30
62tracking	0	23,600	100	23,700	23,700	24
C-ZeroMin	0	25,600	100	25,700	25,700	26
C-25%Min	0	26,300	200	26,500	26,500	27
62n	0	20,200	100	20,300	20,300	20
C-62nAreaMin	0	19,000	100	19,100	19,100	19
C-Title24	0	34,400	200	34,600	34,600	35
Conference Room						
62/2001	17,200	18,400	100	18,500	35,700	357
62tracking	2,400	4,700	0	4,700	7,100	71
62/Int	7,600	9,200	100	9,300	16,900	169
C-ZeroMin	3,900	7,200	0	7,200	11,100	111
C-25%Min	6,000	9,000	100	9,100	15,100	151
62n	3,600	5,700	0	5,700	9,300	93
C-62nAreaMin	300	2,700	0	2,700	3,000	30
C-Title24	4,700	8,200	0	8,200	12,900	129
Lecture Hall						
62/2001	51,800	52,800	300	53,100	104,900	1049
62tracking	18,200	20,000	100	20,100	38,300	383
62/Int	19,800	26,400	200	26,600	46,400	464
C-ZeroMin	26,500	30,100	200	30,300	56,800	568
C-25%Min	30,300	34,000	200	34,200	64,500	645
62n	22,400	28,200	200	28,400	50,800	508
C-62nAreaMin	7,300	16,800	100	16,900	24,200	242
C-Title24	34,400	36,800	200	37,000	71,400	714
Classroom						
62/2001	9,600	10,000	100	10,100	19,700	197
62tracking	2,600	5,400	0	5,500	8,100	81
C-ZeroMin	3,100	6,600	0	6,600	9,700	97
C-25%Min	3,200	7,300	0	7,300	10,500	105
62n	7,800	8,900	100	9,000	16,800	168
C-62nAreaMin	2,300	6,500	0	6,500	8,800	88
C-Title24	3,900	8,000	0	8,000	11,900	119
Portable Classroom						
62/2001	4,700	4,900	0	4,900	9,600	108
62tracking	2,300	3,200	0	3,200	5,500	62
C-ZeroMin	2,200	3,500	0	3,500	5,700	64
C-25%Min	2,500	3,800	0	3,800	6,300	71
62n	4,800	5,000	0	5,000	9,800	110
C-62nAreaMin	2,800	4,000	0	4,000	6,800	76
C-Title24	3,700	4,700	0	4,700	8,400	94
Fast Food						
62/2001	74,200	53,000	400	53,400	127,600	1021
62tracking	18,600	26,400	200	26,600	45,200	362
C-ZeroMin	29,400	34,900	200	35,100	64,500	516
C-25%Min	34,000	37,500	300	37,800	71,800	574
62n	25,600	28,600	200	28,800	54,400	435
C-62nAreaMin	6,600	20,900	200	21,100	27,700	222
C-Title24	27,500	33,500	200	33,700	61,200	490

Table A1 Detailed Energy Loads for Bakersfield

	Heating (MJ)	Sensible cooling (MJ)	Latent cooling (MJ)	Total cooling (MJ)	Total heating & cooling (MJ)	Total heating & cooling (MJ/m^2)
Office						
62/2001	0	2,700	3,100	5,800	5,800	6
62tracking	0	2,300	2,400	4,700	4,700	5
C-ZeroMin	0	2,300	2,300	4,600	4,600	5
C-25%Min	0	2,400	2,500	4,900	4,900	5
62n	0	1,800	2,100	3,900	3,900	4
C-62nAreaMin	0	1,700	1,900	3,600	3,600	4
C-Title24	0	3,100	3,600	6,700	6,700	7
Conference Room						
62/2001	13,600	1,900	1,800	3,700	17,300	173
62tracking	1,700	400	400	800	2,500	25
62/Int	4,500	900	900	1,800	6,300	63
C-ZeroMin	2,900	600	600	1,200	4,100	41
C-25%Min	3,900	900	800	1,700	5,600	56
62n	1,000	600	600	1,200	2,200	22
C-62nAreaMin	0	200	200	400	400	4
C-Title24	3,300	800	700	1,500	4,800	48
Lecture Hall						
62/2001	43,900	4,200	4,700	8,900	52,800	528
62tracking	10,200	2,000	2,000	4,000	14,200	142
62/Int	8,200	2,100	2,400	4,500	12,700	127
C-ZeroMin	16,100	2,800	3,000	5,800	21,900	219
C-25%Min	18,400	3,100	3,300	6,400	24,800	248
62n	10,900	2,300	2,500	4,800	15,700	157
C-62nAreaMin	900	1,500	1,600	3,100	4,000	40
C-Title24	23,200	3,400	3,600	7,000	30,200	302
Classroom						
62/2001	3,500	1000	1,100	2,100	5,600	56
62tracking	400	600	600	1,200	1,600	16
C-ZeroMin	500	700	700	1,400	1,900	19
C-25%Min	600	700	800	1,500	2,100	21
62n	2,400	900	1000	1,900	4,300	43
C-62nAreaMin	200	700	700	1,400	1,600	16
C-Title24	700	800	800	1,600	2,300	23
Portable Classroom						
62/2001	2,100	500	600	1,100	3,200	36
62tracking	700	400	400	800	1,500	17
C-ZeroMin	700	400	400	800	1,500	17
C-25%Min	700	400	400	800	1,500	17
62n	2,200	600	600	1,200	3,400	38
C-62nAreaMin	700	400	400	800	1,500	17
C-Title24	1,400	500	500	1,000	2,400	27
Fast Food						
62/2001	55,100	4,100	5,100	9,200	64,300	514
62tracking	7,200	2,200	2,400	4,600	11,800	94
C-ZeroMin	13,900	2,800	3,100	5,900	19,800	158
C-25%Min	16,100	3,000	3,400	6,400	22,500	180
62n	4,200	2,200	2,800	5,000	9,200	74
C-62nAreaMin	100	1,600	1,900	3,500	3,600	29
C-Title24	10,000	2,600	3,000	5,600	15,600	125

Table A2 Detailed Energy Loads for Los Angeles

	Heating (MJ)	Sensible cooling (MJ)	Latent cooling (MJ)	Total cooling (MJ)	Total heating & cooling (MJ)	Total heating & cooling (MJ/m^2)
Office						
62/2001	0	18,100	100	18,200	18,200	18
62tracking	0	14,900	100	15,000	15,000	15
C-ZeroMin	0	16,400	100	16,500	16,500	17
C-25%Min	0	16,600	100	16,700	16,700	17
62n	0	12,200	100	12,300	12,300	12
C-62nAreaMin	0	11,700	100	11,800	11,800	12
C-Title24	0	20,800	100	20,900	20,900	21
Conference Room						
62/2001	22,900	11,800	100	11,900	34,800	348
62tracking	3,600	3,000	0	3,000	6,600	66
62/Int	10,300	5,900	0	5,900	16,200	162
C-ZeroMin	5,700	4,700	0	4,700	10,400	104
C-25%Min	8,700	5,800	0	5,800	14,500	145
62n	5,200	3,700	0	3,700	8,900	89
C-62nAreaMin	500	1,700	0	1,700	2,200	22
C-Title24	6,900	5,300	0	5,300	12,200	122
Lecture Hall						
62/2001	70,000	30,800	200	31,000	101,000	1010
62tracking	24,600	11,500	100	11,600	36,200	362
62/Int	28,200	15,400	100	15,500	43,700	437
C-ZeroMin	35,900	17,700	100	17,800	53,700	537
C-25%Min	41,300	20,000	100	20,100	61,400	614
62n	31,400	16,400	100	16,500	47,900	479
C-62nAreaMin	11,400	10,000	100	10,100	21,500	215
C-Title24	46,800	21,800	100	21,900	68,700	687
Classroom						
62/2001	13,200	6,200	0	6,200	19,400	194
62tracking	3,900	3,500	0	3,500	7,400	74
C-ZeroMin	4,400	4,300	0	4,300	8,700	87
C-25%Min	4,700	4,600	0	4,600	9,300	93
62n	11,100	5,500	0	5,500	16,600	166
C-62nAreaMin	3,800	4,200	0	4,200	8,000	80
C-Title24	5,700	5,100	0	5,100	10,800	108
Portable Classroom						
62/2001	6,300	3,100	0	3,100	9,400	106
62tracking	3,100	2,000	0	2,000	5,100	57
C-ZeroMin	3,100	2,300	0	2,300	5,400	61
C-25%Min	3,400	2,400	0	2,400	5,800	65
62n	6,500	3,100	0	3,200	9,700	109
C-62nAreaMin	3,900	2,500	0	2,500	6,400	72
C-Title24	5,200	3,000	0	3,000	8,200	92
Fast Food						
62/2001	102,500	27,500	100	27,600	130,100	1041
62tracking	26,500	14,100	100	14,200	40,700	326
C-ZeroMin	42,400	18,600	100	18,700	61,100	489
C-25%Min	48,700	19,900	100	20,000	68,700	550
62n	37,700	14,800	100	14,900	52,600	421
C-62nAreaMin	10,600	11,100	100	11,200	21,800	174
C-Title24	40,200	17,800	100	17,900	58,100	465

Table A3 Detailed Energy Loads for Sacramento

	Heating (MJ)	Sensible cooling (MJ)	Latent cooling (MJ)	Total cooling (MJ)	Total heating & cooling (MJ)	Total heating & cooling (MJ/m^2)
Office						
62/2001	0	1,200	0	1,200	1,200	1
62tracking	0	1,000	0	1,000	1,000	1
C-ZeroMin	0	1,100	0	1,100	1,100	1
C-25%Min	0	1,100	0	1,100	1,100	1
62n	0	800	0	800	800	1
C-62nAreaMin	0	800	0	800	800	1
C-Title24	0	1,400	0	1,400	1,400	1
Conference Room						
62/2001	29,000	800	0	800	29,800	298
62tracking	4,200	200	0	200	4,400	44
62/Int	12,300	400	0	400	12,700	127
C-ZeroMin	7,000	300	0	300	7,300	73
C-25%Min	10,200	400	0	400	10,600	106
62n	5,000	300	0	300	5,300	53
C-62nAreaMin	300	100	0	100	400	4
C-Title24	8,200	300	0	300	8,500	85
Lecture Hall						
62/2001	91,200	1,900	0	1,900	93,100	931
62tracking	28,300	900	0	900	29,200	292
62/Int	31,200	1,000	0	1,000	32,200	322
C-ZeroMin	41,600	1,200	0	1,200	42,800	428
C-25%Min	48,800	1,400	0	1,400	50,200	502
62n	36,200	1,000	0	1,000	37,200	372
C-62nAreaMin	8,800	700	0	700	9,500	95
C-Title24	57,000	1,500	0	1,500	58,500	585
Classroom						
62/2001	12,800	400	0	400	13,200	132
62tracking	3,200	200	0	200	3,400	34
C-ZeroMin	4,100	300	0	300	4,400	44
C-25%Min	4,200	300	0	300	4,500	45
62n	10,100	400	0	400	10,500	105
C-62nAreaMin	3,000	300	0	300	3,300	33
C-Title24	4,900	400	0	400	5,300	53
Portable Classroom						
62/2001	6,800	200	0	200	7,000	79
62tracking	3,000	100	0	100	3,100	35
C-ZeroMin	3,100	200	0	200	3,300	37
C-25%Min	3,400	200	0	200	3,600	40
62n	7,000	200	0	200	7,200	81
C-62nAreaMin	3,700	200	0	200	3,900	44
C-Title24	5,400	200	0	200	5,600	63
Fast Food						
62/2001	125,500	1,800	0	1,800	127,300	1018
62tracking	27,600	1,000	0	1,000	28,600	229
C-ZeroMin	46,300	1,300	0	1,300	47,600	381
C-25%Min	53,800	1,400	0	1,400	55,200	442
62n	30,700	1,000	0	1,000	31,700	254
C-62nAreaMin	5,500	800	0	800	6,300	50
C-Title24	41,900	1,200	0	1,200	43,100	345

Table A4 Detailed Energy Loads for San Francisco

	Heating (MJ)	Sensible cooling (MJ)	Latent cooling (MJ)	Total cooling (MJ)	Total heating & cooling (MJ)	Total heating & cooling (MJ/m^2)
Office						
62/2001	0	37,500	79,200	116,700	116,700	117
62tracking	0	29,200	56,100	85,300	85,300	85
C-ZeroMin	0	30,400	56,900	87,300	87,300	87
C-25%Min	0	31,700	61,500	93,200	93,200	93
62n	0	25,500	53,800	79,300	79,300	79
C-62nAreaMin	0	23,300	47,300	70,600	70,600	71
C-Title24	0	43,300	91,400	134,700	134,700	135
Conference Room						
62/2001	1,600	22,800	42,600	65,400	67,000	670
62tracking	100	5,200	9,400	14,600	14,700	147
62/Int	500	11,400	21,300	32,700	33,200	332
C-ZeroMin	200	8,100	14,500	22,600	22,800	228
C-25%Min	300	10,600	19,400	30,000	30,300	303
62n	200	7,100	13,200	20,300	20,500	205
C-62nAreaMin	0	3,100	5,600	8,700	8,700	87
C-Title24	200	9,400	17,200	24,600	24,800	248
Lecture Hall						
62/2001	4,300	62,300	127,700	190,000	194,300	1943
62tracking	1,400	25,500	52,100	77,600	79,000	790
62/Int	1,200	31,200	63,800	95,000	96,200	962
C-ZeroMin	2,000	37,400	74,900	112,300	114,300	1143
C-25%Min	2,300	41,600	83,900	125,500	127,800	1278
62n	1,200	33,200	68,100	101,300	102,500	1025
C-62nAreaMin	300	20,600	41,100	61,700	62,000	620
C-Title24	2,800	45,500	91,400	136,900	139,700	140
Classroom						
62/2001	700	12,900	27,000	39,900	40,600	406
62tracking	100	7,100	13,100	20,200	20,300	203
C-ZeroMin	200	8,300	15,100	23,400	23,600	236
C-25%Min	200	9,200	17,700	26,900	27,100	271
62n	600	11,600	24,200	35,800	36,400	364
C-62nAreaMin	100	8,300	15,800	24,100	24,200	242
C-Title24	200	10,200	19,600	29,800	30,000	300
Portable Classroom						
62/2001	400	6,500	12,600	19,100	19,500	219
62tracking	100	4,200	7,700	11,900	12,000	135
C-ZeroMin	100	4,500	8,100	12,600	12,700	143
C-25%Min	100	4,900	9,000	13,900	14,000	157
62n	400	6,600	12,800	19,400	19,800	222
C-62nAreaMin	100	5,100	9,600	14,700	14,800	166
C-Title24	200	6,100	11,400	17,500	17,700	199
Fast Food						
62/2001	7,200	65,200	162,000	227,200	234,400	1875
62tracking	1000	31,700	71,400	103,100	104,100	833
C-ZeroMin	1,700	41,600	95,300	136,900	138,600	1109
C-25%Min	2,100	45,200	106,200	151,400	153,500	1228
62n	1,500	35,300	87,600	122,900	124,400	995
C-62nAreaMin	200	25,200	59,500	84,700	84,900	679
C-Title24	1,600	40,300	94,300	134,600	136,200	1090

Table A5 Detailed Energy Loads for Miami

	Heating (MJ)	Sensible cooling (MJ)	Latent cooling (MJ)	Total cooling (MJ)	Total heating & cooling (MJ)	Total heating & cooling (MJ/m^2)
Office						
62/2001	47,100	7,300	8,500	15,800	62,900	63
62tracking	21,900	5,900	6,300	12,200	34,100	34
C-ZeroMin	22,300	6,400	6,600	13,000	45,300	45
C-25%Min	23,000	6,600	7,000	13,600	36,600	37
62n	7,600	5,000	5,800	10,800	18,400	18
C-62nAreaMin	2,700	4,700	5,200	9,900	12,600	13
C-Title24	75,300	8,400	9,800	18,200	93,500	94
Conference Room						
62/2001	63,300	4,600	4,800	9,400	72,700	727
62tracking	12,600	1,100	1,100	2,200	14,800	148
62/Int	30,900	2,300	2,400	4,700	35,600	356
C-ZeroMin	20,000	1,700	1,600	3,300	23,300	233
C-25%Min	27,600	2,200	2,200	4,400	32,000	320
62n	18,400	1,400	1,500	2,900	21,300	213
C-62nAreaMin	5,900	600	600	1,200	7,100	71
C-Title24	24,100	2,000	1,900	3,900	28,000	280
Lecture Hall						
62/2001	189,500	12,800	14,500	27,300	216,800	2168
62tracking	73,500	4,900	5,700	10,600	84,100	841
62/Int	90,400	6,400	7,300	13,700	104,100	1041
C-ZeroMin	107,300	7,400	8,400	15,800	123,100	1231
C-25%Min	121,800	8,300	9,400	17,700	139,500	1395
62n	97,100	6,800	7,700	14,600	111,700	1117
C-62nAreaMin	53,100	4,100	4,600	8,700	61,800	618
C-Title24	132,900	9,000	10,200	19,200	152,100	1521
Classroom						
62/2001	39,300	2,400	2,900	5,300	44,600	446
62tracking	17,300	1,400	1,500	2,900	20,200	202
C-ZeroMin	19,400	1,700	1,700	3,400	22,800	228
C-25%Min	22,700	1,800	1,900	3,700	26,400	264
62n	34,900	2,200	2,600	4,800	39,700	397
C-62nAreaMin	19,700	1,600	1,700	3,300	23,000	230
C-Title24	26,100	2,000	2,200	4,200	30,300	303
Portable Classroom						
62/2001	18,400	1,200	1,400	2,600	21,000	236
62tracking	10,700	800	800	1,600	12,300	138
C-ZeroMin	11,200	900	900	1,800	13,000	146
C-25%Min	12,500	1,000	1000	2,000	14,500	163
62n	18,800	1,200	1,400	2,600	21,400	240
C-62nAreaMin	13,500	1,000	1,100	2,100	15,600	175
C-Title24	16,400	1,200	1,300	2,500	18,900	212
Fast Food						
62/2001	241,200	12,100	18,100	30,200	271,400	2171
62tracking	91,400	6,200	8,300	14,500	105,900	847
C-ZeroMin	131,400	8,100	11,000	19,100	150,500	1204
C-25%Min	148,800	8,700	12,100	20,800	169,600	1357
62n	121,900	6,500	9,800	16,300	138,200	1106
C-62nAreaMin	72,400	4,800	6,800	11,600	84,000	672
C-Title24	128,700	7,800	10,800	18,600	147,300	1178

Table A6 Detailed Energy Loads for Minneapolis

Appendix B: PIER RFP Issues

The California Energy Commission (CEC) Public Interest Energy Research (PIER) Request for Proposals (RFP) for the Buildings Energy Efficiency Program Area identified four broad issues of key concern to the CEC. These four issues identify energy problems facing buildings in California and present opportunities to have a significant positive impact. This appendix will discuss the relationship of the application of CO_2-based DCV systems in small non-residential buildings to the four key issues based on information in this report.

Issue #1 Energy consumption is rapidly increasing in hotter, inland areas as new building construction increases in these areas.

Obviously, the primary intent of CO_2-based DCV systems are to reduce energy consumed to cool or heat ventilation air in buildings and, as demonstrated in this report, they are capable of achieving such reductions for many building types in a variety of climates. These results indicate that CO_2 DCV systems can reduce cooling energy consumption in the hotter, inland areas of California in many occupancies. As application of CO_2 DCV in new construction is considered, some thought will need to be given to the possibility that these newer buildings may have low infiltration rates during unoccupied periods and some strategy may be needed to address contaminant buildup when the system is off.

Issue #2 Development of energy efficient products and services needs to adequately consider non-energy benefits, such as comfort, productivity, durability, and decreased maintenance.

Since CO_2-based DCV systems directly affect ventilation rates provided in buildings, there is the potential to have a significant impact on building occupant comfort and productivity. That impact could be either positive or negative depending on the DCV system design, installation, operation and maintenance. CO_2-based DCV systems can have a positive impact on IAQ that is not commonly considered. Frequently, building zones are occupied by more people than the ventilation system design criteria. At such times, a DCV system will result in improved IAQ by increasing the ventilation supplied to the space. Additionally, ventilation systems may operate with lower ventilation effectiveness than the design criteria. Again, a DCV system can increase ventilation rates in such situations. While it is not possible to estimate potential impacts on productivity for any given building, Fisk and Rosenfeld (1997) have estimated that nationwide impacts of better indoor environments are in the billions of dollars.

Since DCV systems adjust ventilation rates based on measured concentrations of CO_2 generated by building occupants, they do not directly guarantee satisfactory indoor air quality (IAQ) due to the presence of non-occupant generated contaminants. This results in a concern by some that DCV could result in poor IAQ that would negatively impact comfort and productivity. Certain steps need to be taken to avoid the occurrence of such a negative impact. The most fundamental step is to implement the same good IAQ practices that should be applied to all commercial buildings. This includes such practices as reducing contaminant sources, properly installing and maintaining equipment, etc. Additional steps that should be taken for DCV systems include appropriate selection of control algorithms and setpoints, thoughtful consideration of expected contaminant sources, establishing needed minimum base and/or purge ventilation rates and schedules, and proper maintenance and calibration of CO_2 sensors.

Finally, the impacts of DCV systems on comfort and productivity have not been thoroughly studied. Since this is an important issue, more research in this area is needed.

Issue #3 Building design, construction, and operation of energy-related features can affect public health and safety.

The discussion above addressing Issue #2 applies equally to public health. CO_2-based DCV systems could have either a negative or positive impact on public health and care needs to be taken in their application. Specifically for moisture issues, DCV can have a very positive impact in lessening the moisture load in non-residential buildings in humid climates. Since most of the moisture load for many non-residential buildings is brought into a building through ventilation, reducing excess ventilation during times of reduced building occupancy can reduce this moisture load. This reduction in moisture load can save energy and money by eliminating the need for special equipment.

Issue #4 Investments in energy efficiency can affect building and housing affordability and value, and the state's economy.

As discussed in response to Issue #1, CO_2-based DCV systems can reduce building heating and cooling energy use and, therefore, reduce operating costs to improve building affordability and value. However, these potential savings will vary widely depending on building type, climate, occupancy density and patterns, other HVAC equipment used, and other factors. While knowledge of these important parameters is growing, more work is needed to identify the best opportunities for energy savings. No impacts are expected on the energy-related costs of construction.

Fisk, W.J. and A.H. Rosenfeld. Estimates of Improved Productivity and Health from Better Indoor Environments (1997) Indoor Air 1997; 7:158-172.

Recommendations for Application of CO_2-Based Demand Controlled Ventilation, Including Proposed Guidance for ASHRAE Standard 62 and California's Title 24

Letter Report on Task 3.1.5a and 3.1.6a of CEC-EEB RMT Project

Andrew Persily[1], Jim Braun[2], Steve Emmerich[1], Kevin Mercer[2], Tom Lawrence[2]

[1]Building and Fire Research Laboratory
National Institute of Standards and Technology

[2]Ray W. Herrick Laboratories
Purdue University

Abstract
Carbon dioxide (CO_2) based demand controlled ventilation (DCV) has been proposed and implemented for many years as a strategy for increasing energy efficiency by providing outdoor air ventilation rates based on actual occupancy rather than design occupancy. Over the years there have been various simulation studies and demonstration projects, but important questions remain about energy savings potential, indoor air quality impacts, and both when and how to apply the approach. A major element of the CEC-EEB project examined the impact and application of CO_2 DCV via a literature review, field demonstrations, energy analysis, and indoor air quality simulations. Based on the results of these efforts, this report presents a number of recommendations on the application of CO_2 DCV.

1. Introduction

Demand controlled ventilation (DCV) is a ventilation rate control strategy to address the concern that when a space is occupied at less than its design occupancy, unnecessary energy consumption can result if the space is ventilated at the design outdoor air rate rather than the ventilation rate based on the actual occupancy. Furthermore, early during a given day of building occupancy, contaminants generated by people and their activities will not yet have reached their ultimate levels based on the transient nature of contaminant buildup. As a result, it may be possible to delay or lag the onset of the design ventilation rate to take credit for this transient effect. A number of approaches have been proposed to account for actual occupancy levels and to provide for ventilation corresponding to the actual rather than design occupancy. These include time-based scheduling when occupancy patterns are well-known and predictable, occupancy sensors to determine when people have entered a space (though not necessarily how many), and carbon dioxide (CO_2) sensing and control as a means of estimating the number of people in a space or the strength of occupant-related contaminant sources.

Controlling outdoor air intake rates using CO_2 DCV offers the possibility of reducing the energy penalty of over-ventilation during periods of low occupancy, while still ensuring adequate levels of outdoor air ventilation. In addition, CO_2 DCV gives credit for building ventilation due to infiltration through the building envelope, which can be significant even in mechanically ventilated buildings. A number of studies have identified the potential energy savings of CO_2 DCV in commercial and institutional buildings via field studies and computer simulations, and have shed light on the magnitude of energy savings possible and the dependence of these savings on climate, building and system type, control approach, and occupancy patterns. However, important issues remain to be resolved in the application of CO_2 DCV including how best to apply the approach, which in turn includes issues such as which control algorithm to use in a given building, sensor location, sensor maintenance and calibration, and the amount of baseline ventilation required to control contaminant sources that do not depend on the number of occupants. This report presents application guidance based on the CEC-EEB project on CO_2 DCV carried out by the National Institute of Standards and Technology (NIST) and Purdue University.

2. Recommendations Based on Previously Published Literature

An earlier task under this project involved a literature review on the subject of CO_2-based DCV (Emmerich and Persily 2001). Some of this literature contained guidance on the application of this ventilation control approach, which was summarized in an appendix to the literature review report. An updated version of this guidance is presented in this section organized into sections on target buildings where CO_2-based DCV appears to be most applicable, CO_2 sensor technology issues, ventilation control algorithms, control of contaminants not associated with occupants, and other considerations.

<u>Target buildings</u>

The intent of CO_2-based DCV is to reduce energy use compared to ventilation at a constant rate based on design occupancy, while assuring adequate ventilation rates for IAQ control. While CO_2-based DCV systems are likely to save at least some energy in nearly all buildings and climates, the amount of energy saved can vary dramatically depending on the climate, occupancy, operating hours, and other building and HVAC system features. The greatest energy savings are likely to occur in buildings with large heating loads and/or large cooling loads that have dense occupancies that vary unpredictably. Even though a building as a whole might not be a good candidate, CO_2 DCV may be appropriate within specific spaces in such a building that have independent outdoor air supply capability, such as a conference rooms.

CO_2 DCV is less likely to be applicable in buildings or spaces where non-occupant generated pollutants dominate ventilation requirements or where there are significant sources of CO_2 other than occupants. Using CO_2 as the control variable in such applications will not necessarily result in unacceptable IAQ but rather could lead to excessive ventilation rates. Buildings or spaces with CO_2 removal mechanisms other than ventilation would similarly not be good candidates. However, such removal mechanisms and non-occupant CO_2 sources are unlikely to exist in most commercial and institutional buildings.

<u>CO_2 DCV Technology</u>

Most CO_2 sensors used in DCV systems today are based on non-dispersive infrared (NDIR) or photometric detection, both of which can be affected by light source aging. The former approach may also be sensitive to particle buildup on the sensor, while the latter could be affected by vibration or atmospheric pressure changes. In selecting sensors for ventilation control, one needs to consider the appropriate measurement ranges for ventilation control, approximately 540 mg/m^3 (300 ppm(v)) to 2700 mg/m^3 (1500 ppm(v)). The sensors employed also need to be calibrated and maintained according to manufacturer recommendations. Some manufacturers have proposed automated calibration checks using overnight baseline CO_2 readings.

In locating CO_2 sensors for ventilation control, one should avoid locations near doors, windows, air intakes or exhausts, or in close proximity to occupants. Also, a single sensor located in a common return should not generally be used to control ventilation rates for multiple spaces with different occupancies.

<u>Control Algorithms</u>

Control strategies for CO_2-based DCV include two-position (on-off) control, setpoint simple control where the ventilation rate is increased or decreased depending on the indoor CO_2

concentration, proportional control in which the ventilation rate is proportional to the CO_2 concentration, PI (proportional-integral) or PID (proportional-integral-derivative) control which can adjust more quickly and in a more stable manner to changes in the CO_2 concentration. Control strategies should be chosen based on the expected occupancy patterns.

Non-occupant Contaminants

CO_2-based DCV systems should include a strategy to provide for sufficient ventilation, or other means (e.g. reducing contaminant emissions, local ventilation and air cleaning), to control concentrations of non-occupant generated contaminants. Ideally, an analysis of non-occupant sources would indicate the appropriate ventilation rates and other IAQ control technologies needed to maintain the resulting concentrations of contaminants within acceptable limits. However, the information needed to perform such an analysis, primarily contaminant emission rates and air cleaning system efficiencies, are not available in all situations.

Other Considerations

The selection and design of a CO_2-based DCV system cannot be viewed in isolation, as the air quality and energy performance will impact and be impacted by other building and HVAC systems. While the interactions will be building and system specific, interactions can occur with economizers, displacement ventilation, and other technologies. For example, in buildings with an economizer cycle, the economizer should be allowed to override the DCV system at times when the additional ventilation would provide 'free' cooling. For buildings dominated by cooling loads, DCV should not be used in most climates without an economizer. For buildings with displacement ventilation, in which conditions are created such that both air temperatures and contaminant concentrations are stratified, CO_2 sensors should be located within the occupants' breathing zone or the control system should somehow account for concentration gradients due to such stratification.

Also, installation of an outdoor CO_2 sensor should be considered if outdoor levels are expected to vary significantly over time or to deviate significantly (more than about 20 %) from 720 mg/m^3 (400 ppm(v)). The outdoor CO_2 concentrations can be assumed to be 720 mg/m^3 (400 ppm(v)) for most applications, but urban areas may have local effects resulting in higher levels. The higher outdoor level could result in overventilation and it may be economical to install an additional sensor to control the difference between indoor and outdoor concentration directly. Such an installation may also be required by some applicable standards or codes.

3. Recommendations Based on Project Technical Work

Other phases of this project involved technical work by both NIST and Purdue University to examine different aspects of CO_2 based DCV. NIST performed simulations of CO_2 DCV performance in a variety of space types for several different ventilation control approaches (Persily et al. 2003). The results of these simulations were compared in terms of ventilation rates, indoor CO_2 levels, indoor concentrations of a generic VOC (volatile organic compound) contaminant intended to represent non-occupant contaminant sources, and the energy consumption associated with ventilation. Purdue University performed a detailed economic assessment of CO_2 DCV for a range of different building types and climates within California (Braun et al. 2003). In this effort, the economics of CO_2 DCV were compared with competing energy recovery technologies, including enthalpy exchanger heat recovery (HXHR), and heat pump heat recovery (HPHR) using both simulation and field studies. As a result of these efforts, new insights were obtained into the application of CO_2 DCV, which are summarized below.

Recommendations Based on NIST Simulation Effort

The primary finding of the NIST simulation effort is that CO_2 DCV is capable of providing acceptable control of indoor contaminants from both typical occupant and non-occupant sources. However, the results also indicate that the performance depends on the details of the spaces including occupancy patterns, ventilation rate requirements in the relevant standard and ventilation system operating schedule as well as the other values used in the analysis, specifically contaminant source strengths and system-off infiltration rates. Among the findings with implications for the application of CO_2 based DCV are the following:

For some space types, CO_2 DCV has the potential to save a large amount of energy. Characteristics of such spaces are highly variable occupancies with high occupant densities at peak occupancy. Examples of such space types include lecture halls, conferences rooms, and classrooms.

For other space types, the potential energy savings are far more climate specific, and the potential savings in mild climates may be insignificant. Characteristics of such spaces are constant or moderately variable occupancies with low peak occupant densities. An example of this type of space is a typical office, but savings with DCV systems may still be possible in more severe climates.

A nonzero minimum or base ventilation rate should be maintained to handle non-occupant sources. The simulations used a value that was 25 % of the design ventilation rate, and this value maintained indoor contaminant levels close to those seen in an idealized system in which the outdoor air intake tracked occupancy perfectly. Simulations with a minimum ventilation rate of zero resulted in elevated contaminant levels, particularly in the early morning. A minimum below 25 % may be acceptable but was not examined in this study.

In order to deal with the potential for elevated concentrations in the morning, even with a nonzero minimum ventilation rate, DCV systems (and perhaps non-DCV systems as well) should have the capability for increased outdoor air intake before the building is occupied. Sometimes referred to as pre-occupancy "flushout," this capability can help alleviate contaminant overnight buildup while the system is off. The need for such flushing, and the corresponding airflow rate, depends on the contaminant source strengths and the fan-off infiltration rate, both of which are difficult to determine. For odorous sources, early morning odor will provide an indication of the need for a flushing cycle. For non-odorous sources, which can still be a serous concern, the need for a flushing cycle is much more difficult to determine.

When selecting appropriate setpoints for a CO_2 DCV system, they need to be low enough to provide adequate ventilation but high enough to achieve some energy savings. The approach used in this study was to set the upper limit based on the steady-state CO_2 concentration expected at design occupancy and a lower limit about 90 mg/m^3 (50 ppm(v)) above outdoors to avoid the system turning on and off too often. Other approaches to determining these setpoints may also work.

Given the availability of the software tool used in this analysis, and the relative simplicity of the simulations, designers should consider performing similar analyses as part of the design process to examine the impact of various design parameters: setpoints, minimum ventilation rates and operating schedules.

Recommendations Based on Purdue Simulation and Experimental Effort

Compared to competing energy recovery technologies, demand-controlled ventilation coupled with an economizer (DCV+EC) gives the largest cost savings and best economics relative to economizer-only systems for small commercial buildings in California. The greatest cost savings and lowest payback periods occur for buildings that have low average occupancy relative to their peak occupancy, such as auditoriums, gyms and retail stores. From a climate perspective, the greatest savings and lowest payback periods occur in extreme climates (either hot or cold). Mild coastal climates have smaller savings and longer payback periods. In most cases, the payback periods associated with DCV+EC are less than two years.

The savings and trends determined through simulation for DCV were verified through field testing in a number of sites. Field sites were established for three different building types in two different climate zones within California. The building types included: 1) a play area for a fast food restaurant, 2) modular school rooms, and 3) a drug store. In each case, nearly duplicate test buildings were identified in both coastal and inland climate areas. For cooling, greater energy and cost savings were achieved for the restaurant play area and drug store than for the modular schoolrooms. Primarily, this is because these buildings have more variability in their occupancy than the schoolrooms. The largest energy and cost savings were achieved in the hotter, inland climates. The payback period for the inland drug store is less than a year and about 3 years for the inland fast food restaurant play area.

There were no substantial cooling season savings for the modular school rooms. The occupancy for the schools is relatively high with relatively small variability. The school sites are also on timers or controllable thermostats that mean the HVAC units only operate during the normal school day. The schools are also generally unoccupied during the heaviest load portion of the cooling season. Furthermore, the results imply that the average metabolic rate of the students may be higher than the value used in ASHRAE Standard 62-2001 to establish a fixed ventilation rate. The DCV control resulted in lower CO_2 concentrations than for fixed ventilation rate at the modular schoolroom sites in Sacramento.

From an economic perspective, CO_2 DCV with an economizer is the recommended ventilation strategy for most small commercial buildings in California.

4. Potential Revisions to ASHRAE Standard 62 and California Title 24

The latest versions of ASHRAE Standard 62 and Title 24 allow the use of CO_2 DCV. However, based on the results of this project and other recent efforts, some modifications of both documents merit consideration. This section contains initial proposals for such revisions.

ASHRAE Standard 62

Given the structure of ASHRAE Standard 62, including current proposals for revising the standard, the following modifications of the standard should be considered:

- Add a definition of demand controlled ventilation to Section 3 Definitions.
- Add requirements to Section 5 Systems and Equipment that must be met when using CO_2 DCV that address the control capabilities and the CO_2 sensors themselves
- Add requirements to Section 6 Procedures on ventilation rate design procedures when using CO_2 DCV.
- Add requirements to Section 8 Operations and Maintenance for components of CO_2 DCV systems.

A proposed definition of demand controlled ventilation is as follows:

> a ventilation control strategy in which the outdoor air intake rate provided by a system is varied based on actual occupancy of the spaces served by that system rather than on fixed occupancy levels for those spaces. Actual occupancy may be determined directly using occupancy sensors or some other means or indirectly based on established schedules or a surrogate for occupancy such as indoor carbon dioxide concentrations.

The material relevant to CO_2 DCV in the Systems and Equipment section should cover the "hardware" requirements that must be met when employing this ventilation approach. This could be accomplished through a new subsection, which could be based on the following proposed language.

5.x CO_2-Based Demand Controlled Ventilation: Outdoor air ventilation may be controlled based on indoor CO_2 concentration when building occupants are the only significant indoor source of CO_2. When outdoor air ventilation is to be controlled based on indoor CO_2 concentration, the system shall meet the following requirements:

5.x.1 Controls and system components shall be provided to automatically modulate the amount of outdoor air intake based on the output of one or more CO_2 sensors.

5.x.2 CO_2 sensors shall be located in each space served by the system being controlled using CO_2-based demand controlled ventilation. **Exception**: Multiple spaces of the same type per Table 2 (Table 6.1 in Addendum 62n) and with similar load, occupancy patterns and outdoor air fraction may share a sensor.

5.x.3 An outdoor CO_2 sensor shall be included in the control system unless it can be shown that the outdoor CO_2 concentrations are relatively stable, in which case the outdoor value may be assumed to be constant at the lower end of the expected range at the building site.

5.x.4 The CO_2 sensors employed shall have an accuracy of at +/- 90 mg/m^3 (50 ppm(v)) or less.

Section 6 on design procedures is a more complex issue with respect to CO_2 DCV. One perspective is that the section already contains ventilation requirements on a per person basis that still need to be met, even if the number of people by which these requirements are multiplied is varied during operation. The 2001 version of the standard does not speak to a minimum ventilation rate when there are no occupants, but the recently approved addendum 62n actually contains requirements on a per person basis and on a per unit floor area basis (Persily 2001). Under normal conditions, one multiplies the number of people by the per person requirement and the floor area by the per floor area requirement and then adds the two products to determine the total ventilation requirement for the space. When applying CO_2 DCV, one would simply use the floor area requirement for the minimum ventilation requirement. Another potential approach to CO_2 DCV in Section 6 is to describe exactly how one would implement the control approach for different system types, but this would require a fairly lengthy description to cover all system types and circumstances and might best be left to application manuals.

Section 8 of Standard 62 deals with Operations and Maintenance issues. It covers sensors in section 8.4.1.7 and in Table 8-1, nothing that their accuracy needs to be verified every 6 months or as required by an O&M manual identified elsewhere in the standard. Section 8.4.1.7 specifically refers to sensors used in demand controlled ventilation, and therefore CO_2 DCV sensors would be covered. However, it may be worth considering the identification of CO_2 sensors specifically in these sections of the standard.

California Title 24

The California Building Energy Standards AB970 (CEC 2001), also referred to as Title 24, already contains requirements for DCV systems in section 121(c). The most relevant requirements in the 2001 version include the following:

Outdoor air intake rates may be reduced to 0.75 L/s•m^2 (0.15 cfm/ft^2) of conditioned floor area if DCV is used.

When DCV is based on CO_2 levels, the indoor CO_2 must be no more than 1280 mg/m^3 (800 ppm(v)) when the space is occupied, unless the ventilation rate to the space is 7.5 L/s (15 cfm) per person or meets L/s•m^2 (cfm/ft^2) requirements for selected space types.

Locate sensors in the space or the return airstream from the space with no less than one sensor every 2500 m^2 (25,000 ft^2), unless the manufacturer recommends more dense sensor placement.

Based on the results of the simulations performed by NIST, the minimum ventilation rate of 0.75 L/s•m^2 (0.15 cfm/ft^2) is probably higher than necessary, as it precludes the use of DCV in offices. A revision of Title24 should consider the floor area requirements in Addendum 62n to ASHRAE Standard 62 as replacement values for these minimum requirements.

5. Remaining Issues

Previous research efforts, as well as those conducted under the NIST and Purdue projects, have provided much useful information and guidance on the application of CO_2 DCV. However, some important questions remain. In particular, the reliability of CO_2 sensors and other control hardware has not been proven through long-term field testing. For example economizer controls used with packaged air conditioning equipment can sometimes be unreliable, and the use of DCV with such systems adds hardware and new reliability issues. Automated diagnostics for DCV applications may be worth considering in future investigation. Additional questions include: sensor performance versus cost, control algorithm requirements as a function of system and occupancy, baseline minimum ventilation rates for different applications, and appropriate methods for pre-occupancy flushing of building.

6. Acknowledgements

This report was supported by the California Energy Commission through the Architectural Energy Corporation (AEC). The authors acknowledge the contributions of Vern Smith of AEC.

7. Disclaimer

This report was prepared as a result of work sponsored by the California Energy Commission (Commission). It does not necessarily represent the views of the Commission, its employees, or the State of California. The Commission, the State of California, its employees, contractors, and subcontractors make no warranty, express or implied, and assume no legal liability for the information in this report; nor does any party represent that the use of this information will not infringe upon privately owned rights. This report has not been approved or disapproved by the Commission nor has the Commission passed upon the accuracy or adequacy of the information in this report.

8. References

ASHRAE. 2001. Standard 62-2001, Ventilation for Acceptable Indoor Air Quality. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Braun, J.E., K. Mercer, and T. Lawrence. 2003. Evaluation Of Demand Controlled Ventilation, Heat Pump Heat Recovery And Enthalpy Exchangers, HL 2003-10.

California Energy Commission. 2001. AB970 Energy Efficiency Standards for Residential and Nonresidential Buildings, P400-01-024.

Emmerich, S.J. and A.K. Persily. 2001. State-of-the-Art Review of CO_2 Demand Controlled Ventilation Technology and Application. NISTIR 6729, National Institute of Standards and Technology.

Persily, A.K., A. Musser, Emmerich, S.J. and M. Taylor. 2003. Simulations of Indoor Air Quality and Ventilation Impacts of Demand Controlled Ventilation in Commercial and Institutional Buildings. NISTIR 7042, National Institute of Standards and Technology.

Persily, AK. 2001. Addendum 62n: Revising the Ventilation Rate Procedure. ASHRAE Journal 43 (9): 12-13.

EVALUATION OF DEMAND CONTROLLED VENTILATION, HEAT PUMP HEAT RECOVERY AND ENTHALPY EXCHANGERS

Submitted to

California Energy Commission

As a Final Report for Deliverables 3.1.6b and 4.2.6a

Prepared by

James E. Braun, Kevin Mercer, and Tom Lawrence
Purdue University

August 2003

ACKNOWLEDGEMENTS

The research that resulted in this document was part of the "Energy Efficient and Affordable Small Commercial and Residential Buildings Research Program", which was a Public Interest Energy Research Program (PIER) sponsored by the California Energy Commission (CEC). We are extremely grateful for the funding from the CEC through PIER. Several organizations and people were involved in the project described in this final report and deserve our many thanks. Chris Scruton of the CEC was very enthusiastic throughout this project and provided valuable insight and encouragement. Vern Smith and Erin Coats of Architectural Energy Corporation (AEC) provided excellent overall project management for the numerous projects within this large program. Todd Rossi and Doug Dietrich of Field Diagnostic Services, Inc. (FDSI) and Lanny Ross of Newport Design Consultants (NDS) were essential in setting up and maintaining the field sites. Martin Nankin of FDSI provided monitoring equipment at cost for this project. Bob Sundberg of Honeywell, Inc. arranged the donation of Honeywell equipment and was an enthusiastic supporter of the research. Adrienne Thomle and Bill Bray at Honeywell, Inc. made sure that equipment was delivered to the field sites and provided replacement equipment and troubleshooting when necessary. We also had excellent support from the organizations that own and/or operate buildings at the field sites. Individual contributors include Jim Landberg for the Woodland schools, Tadashi Nakadegawa for the Oakland schools, Tony Spata and Mike Godlove for McDonalds, and Mike Sheldon, Paul Hamann, Gordon Pellegrinetti, and Tim Schmid for Walgreens.

EXECUTIVE SUMMARY

The overall objective of the work described in this report was to provide an economic assessment of three alternative ventilation strategies for small commercial buildings in the state of California. The three alternative technologies considered were demand-controlled ventilation (DCV), enthalpy exchanger heat recovery (HXHR), and heat pump heat recovery (HPHR). These three technologies were compared with a base case incorporating fixed ventilation with a differential enthalpy economizer.

The primary evaluation approach involved the use of detailed simulations to estimate operating costs and economic payback periods. A simulation tool, termed the **V**entilation **S**trategy **A**ssessment **T**ool (VSAT), was developed to estimate cost savings associated with the three different ventilation strategies for a set of prototypical buildings and equipment. The buildings considered within VSAT cover a wide range of occupancy schedules and include a small office building, a sit-down restaurant, a retail store, a school class wing, a school auditorium, a school gymnasium, and a school library.

Field sites were also established for the DCV and heat pump heat recovery systems. The goals of the field testing were to verify savings and identify practical problems associated with these technologies. Several field sites were established for DCV that would allow side-by-side testing for different building types in different climates. A single field site was established for the heat pump heat recovery unit in order to verify the performance of the unit.

The simulation study considered both retrofit and new building designs. In both cases, demand-controlled ventilation coupled with an economizer (DCV+EC) was found to give the largest cost savings and best economics relative to an economizer only system for the different prototypical buildings and systems evaluated in the California climate zones. DCV reduces ventilation requirements and loads whenever the economizer is not enabled and the occupancy is less than the peak design value typically used to establish fixed ventilation rates according to ASHRAE Standard 62-1999. Lower ventilation loads lead to lower equipment loads, energy usage and peak electrical demand.

Figure A shows sample payback periods for DCV+EC compared to the base case for a retrofit analysis. The greatest cost savings and lowest payback periods occur for buildings that have low average occupancy relative to their peak occupancy, such as auditoriums, gyms and retail stores. From a climate perspective, the greatest savings and lowest payback periods occur in extreme climates (either hot or cold). The mild coastal climates have smaller savings and longer payback periods. In most cases, the payback period associated with DCV+EC was less than 2 years.

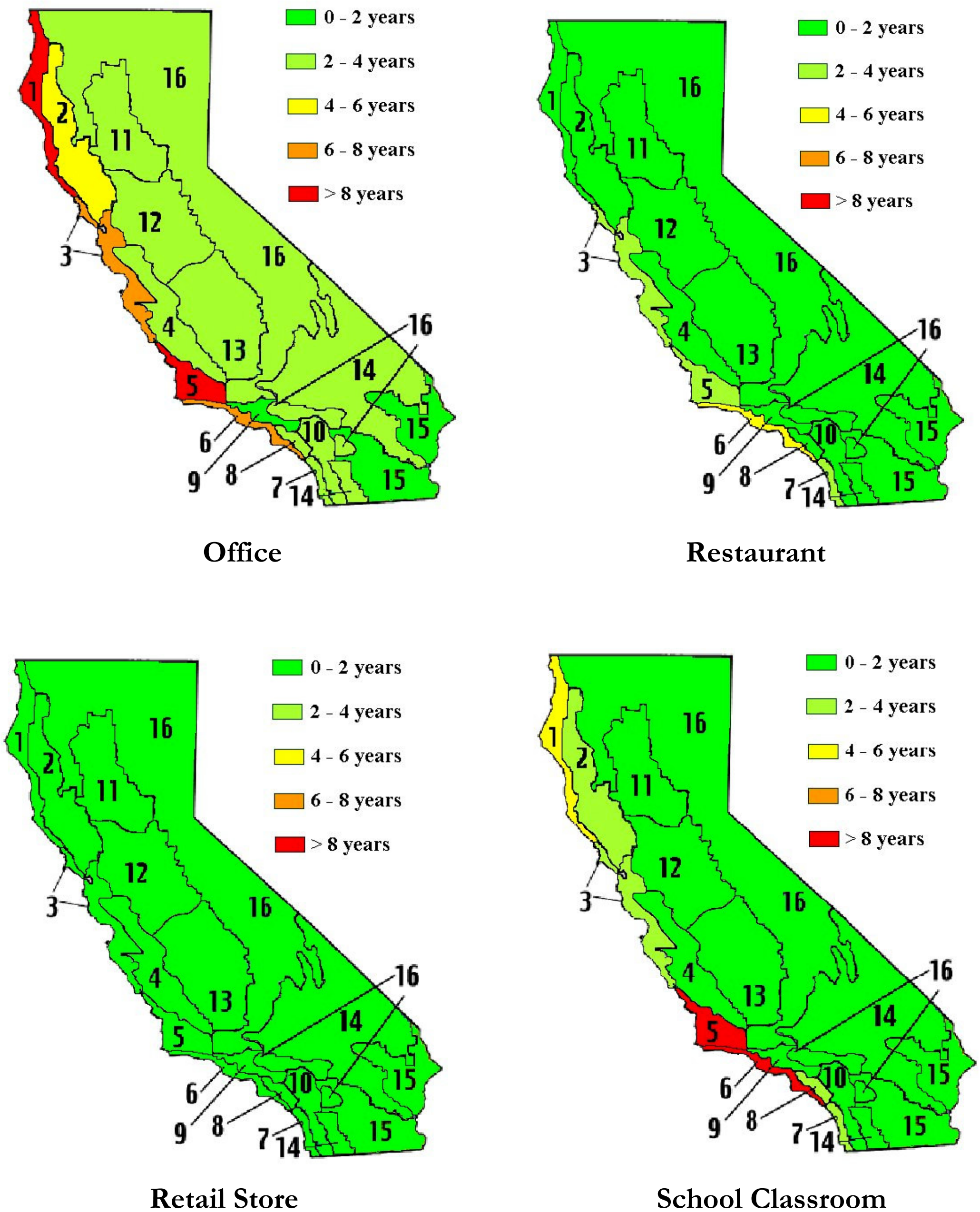


Figure A. Sample Payback Periods for DCV + EC in a Retrofit Application

The heat pump heat recovery (HPHR) system did not provide positive cost savings for many situations investigated for California climates. Heating requirements are relatively low for California climates and therefore overall savings are dictated by cooling season performance. The cooling COP of the HPHR system must be high enough to overcome additional cycling losses from the primary air conditioner compressor, additional fan power associated with the exhaust and/or ventilation fan, additional cooling requirements due to a higher latent removal and a lower operating COP for the primary air conditioner compressor because of a colder mixed air temperature. In addition, the HPHR system is an alternative to an economizer and so economizer savings are also lost when utilizing this system. There are not sufficient hours of ambient temperatures above the breakeven points to yield overall positive savings with the HPHR system compared to a base case system with an economizer for the prototypical buildings in California climates.

The breakeven ambient temperatures for positive savings with the HXHR system are much lower than for the HPHR system because energy recovery (and reduced ventilation load) does not require additional compressor power. The primary penalty is associated with increased fan power due to an additional exhaust fan. In addition, as with the HPHR system, the HXHR system is an alternative to an economizer. Therefore, economizer savings are also lost when utilizing this system. Although positive savings were realized for a number of different buildings and climate zones, the HXHR system had greater operating costs than the DCV system for all cases considered. Furthermore, the initial cost for an HXHR system is higher than a DCV system and also requires higher maintenance costs. Payback for the enthalpy exchanger was found to be greater than 7 years for most all areas of California, except for some building types in climate zone 15.

The payback periods presented in Figure A were calculated assuming a retrofit application. The use of an enthalpy exchanger or heat pump heat recovery unit would lead to a smaller design load for the HVAC equipment which impacts the overall economics. This effect was also considered through simulation. Figures B and C show cumulative rates of return for two different buildings in CACZ 15 as a function of year after the retrofit. The rate of return is the total savings in costs (including a reduction in primary equipment costs) divided by the cost of the ventilation strategy and expressed as a percent. The simple payback period occurs at the point where the rate of return becomes positive. The enthalpy exchanger results in an immediate rate of return (immediate payback) due to RTU equipment cost savings. Although the rates return for the DCV+EC start out negative (due to the initial investment), they surpass the enthalpy exchanger rates of return within a short time period. In general, the rates of return are higher in hotter climates and for the buildings having higher peak occupancy (e.g, the retail store versus the office). Rates of return for both the HXHR and HPHR systems were negative in the moderate climates, but economics for DCV+EC were still positive. In general, the HPHR system is not competitive with the other technologies.

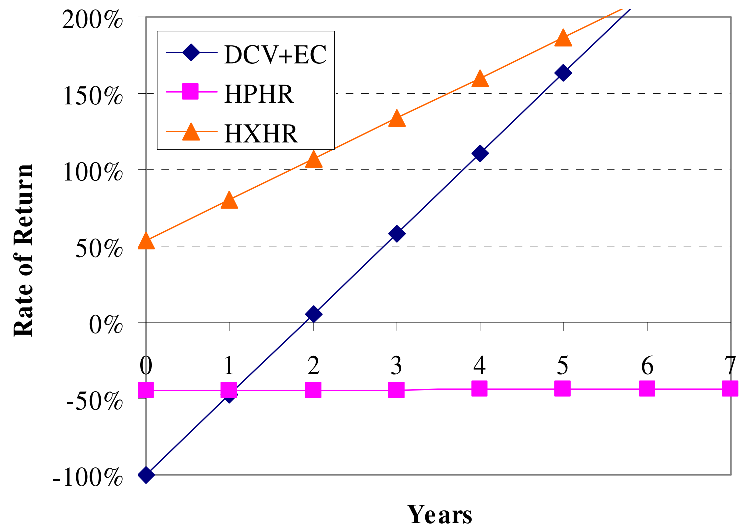


Figure B. Cumulative Rate of Return for New Office Building Design in CACZ 15

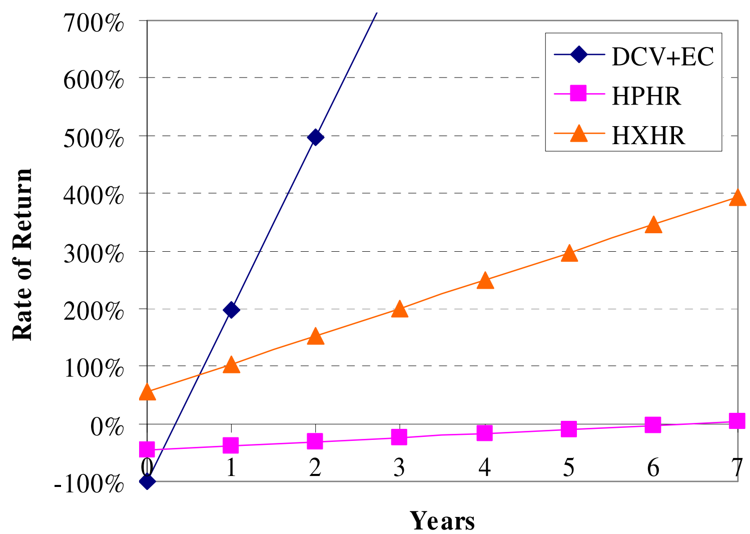


Figure C. Cumulative Rate of Return for New Retail Store in CACZ 15

The different ventilation strategies also have some different effects on comfort conditions due to variations in humidity conditions. For humid climates (outside of California), the alternative ventilation strategies provide lower zone humidity levels than a conventional system during the cooling season. DCV typically provides the lowest zone humidities, followed by the HXHR system, and then the HPHR system.

The savings and trends determined through simulation for DCV were verified through field testing in a number of sites. Field sites were established for three different building types in two different climate zones within California. The building types are: 1)

McDonalds PlayPlace® areas, 2) modular school rooms, and 3) Walgreens drug stores. In each case, nearly duplicate test buildings were identified in both coastal and inland climate areas. For cooling, greater energy and cost savings were achieved at the McDonalds PlayPlaces and Walgreens than for the modular schoolrooms. Primarily, this is because these buildings have more variability in their occupancy than the schoolrooms. The largest energy and cost savings were achieved at the Walgreens in Rialto, followed by the Bradshaw McDonalds PlayPlaces. The Rialto Walgreens appears to have the lowest occupancy and is located in a relatively hot climate with relatively large ventilation loads. The Bradshaw McDonalds PlacePlace appears to have the lowest average occupancy level compared to the other McDonalds PlacePlaces. This site is located in Sacramento and has larger ventilation and total cooling loads than the bay area McDonalds. The payback period for the Rialto Walgreens is less than a year and is between 3 and 6 years for the McDonalds PlayPlaces.

There were no substantial cooling season savings for the modular school rooms. The occupancy for the schools is relatively high with relatively small variability. The school sites are also on timers or controllable thermostats that mean the HVAC units only operate during the normal school day. The schools are also generally unoccupied during the heaviest load portion of the cooling season. Furthermore, the results imply that the average metabolic rate of the students may be higher than the value used in ASHRAE Standard 62-1999 to establish a fixed ventilation rate. In fact, the DCV control resulted in lower CO_2 concentrations than for fixed ventilation rate at the modular schoolroom sites in Sacramento.

A single field site was established for the heat pump heat recovery unit for school in Woodland, CA. The field data confirmed that the steady-state performance of the heat pump in the field is very close to the performance determined in the laboratory and published by the manufacturer for both cooling and heating modes. Furthermore, the model implemented within VSAT for the heat pump accurately predicts capacity and compressor power when compared to recorded field data for steady-state conditions.

For most all locations throughout the state of California, demand-controlled ventilation with an economizer is the recommended ventilation strategy. An enthalpy exchanger is viable in many situations, but DCV was found to have better overall economics for retrofit applications. Heat pump heat recovery is not recommended for California. This technology would make more sense in cold climates where heating costs are more significant. The savings potential for all ventilation strategies is greater in cold climates where heating dominates.

TABLE OF CONTENTS

I. INTRODUCTION

This report describes an assessment of three competing ventilation strategies for reducing ventilation loads in small commercial buildings located in California that utilize packaged equipment. Figure 1 illustrates a typical HVAC system application for small commercial buildings that was considered. A single packaged unit (e.g., a rooftop unit) serves a single zone and incorporates a direct expansion air conditioner, gas or electric heater, a supply fan, and a ventilation system.

The ventilation and exhaust air streams are outlined in Figure 1 to depict the portion of the system where alternative ventilation strategies are employed. The three alternative technologies considered were demand-controlled ventilation, enthalpy exchangers, and heat pump heat recovery. These three technologies were compared with a base case incorporating fixed ventilation with a differential enthalpy economizer.

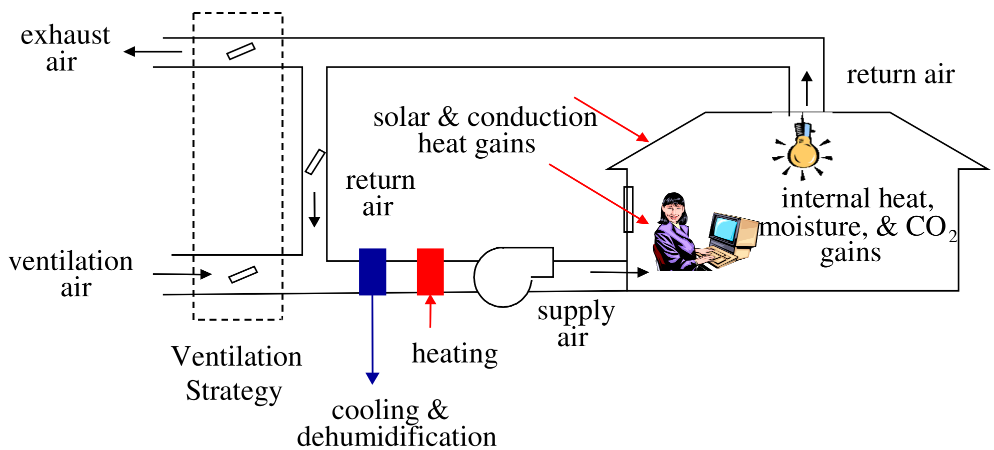


Figure 1. Small Commercial HVAC System

Base Case Ventilation Strategy

The base case ventilation system employs a controllable ventilation and exhaust (and possibly return) damper and a differential enthalpy economizer. The minimum ventilation air flowrate is determined from ASHRAE Standard 62-1999 based upon a design occupancy. The economizer is enabled whenever the ambient enthalpy is less than the return air enthalpy and there is a call for cooling. Under economizer operation, the dampers are controlled to maintain a mixed air temperature set point (e.g., 55 F). With a controllable return damper, this control strategy leads to the use of 100% outside air at many ambient conditions when cooling is required.

Demand-Controlled Ventilation (DCV)

Demand-controlled ventilation involves adjusting the outdoor air ventilation flowrates to maintain a fixed set point for indoor carbon dioxide (CO_2) concentration. The sensor can be placed in the zone or in the return duct. In this study, DCV was considered in combination with an enthalpy economizer. In effect, the minimum ventilation flowrate is determined by the DCV control and the economizer acts to override this minimum and provide additional ventilation flow and a load reduction. During the cooling season, DCV reduces the cooling requirements for the primary equipment whenever there is a call for cooling and the ambient enthalpy is greater than the return air enthalpy. During the heating season, DCV reduces the heating requirements for the primary equipment whenever there is a call for heating and the ambient enthalpy is less than the return air enthalpy. Greater ventilation loads and therefore greater savings opportunities for DCV occur in more extreme climates (hot or cold).

Enthalpy Exchanger Heat Recovery (HXHR)

Figure 2 depicts a typical rotary air-to-air enthalpy exchanger. This device is composed of a revolving cylinder filled with an air-permeable medium having a large internal surface area that transfers both heat and moisture between two air streams. The media is typically fabricated from metal, mineral or polymer materials. The heat and moisture transfer occur between the ventilation and exhaust air streams shown in Figure 2. In the cooling season, an enthalpy wheel can precool and dehumidify the ventilation air reducing the load on the primary air conditioning equipment. In the heating season, the ventilation air is typically preheated and humdified. Greater potential for heat recovery occurs in more extreme climates (hot or cold) because of larger temperature and humidity differences between the ventilation and exhaust air streams. Systems with enthalpy exchangers do not typically incorporate controllable dampers and economizers capable of 100% outside air. The ventilation flow is fixed based upon requirements determined using ASHRAE 62-1999. The wheel is usually controlled based upon the ambient temperature. The wheel rotates when the ambient temperature is either above the return air temperature (cooling) or below a temperature where cooling is not expected (e.g., 55 F). Enthalpy exchangers require an additional exhaust fan to overcome the additional pressure drop associated with flow through the heat exchanger media.

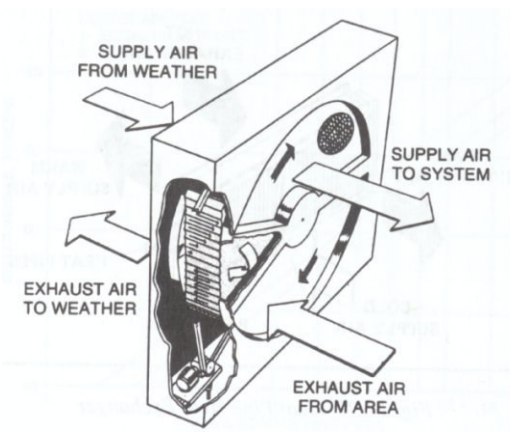


Figure 2. Enthalpy Exchanger (HXHR)

Heat Pump Heat Recovery (HPHR)

Figure 3 shows a heat pump operating between the ventilation and exhaust air streams to recover energy. During the cooling season, the heat pump cools and possibly dehumidifies the ventilation air and rejects heat to the exhaust stream. During the heating season, the heat pump operates in reverse to extract heat from the exhaust air and preheat the outside air. The advantage of this type of system is that the heat pump operates under very favorable conditions as compared with a heat pump having the ambient as a source (heating) or sink (cooling). The COP of the heat pump for heating improves as the ambient gets colder. Similarly, the COP for cooling improves as the ambient gets hotter. Therefore, the savings opportunities for heat pump heat recovery are better in more extreme (hot or cold) climates.

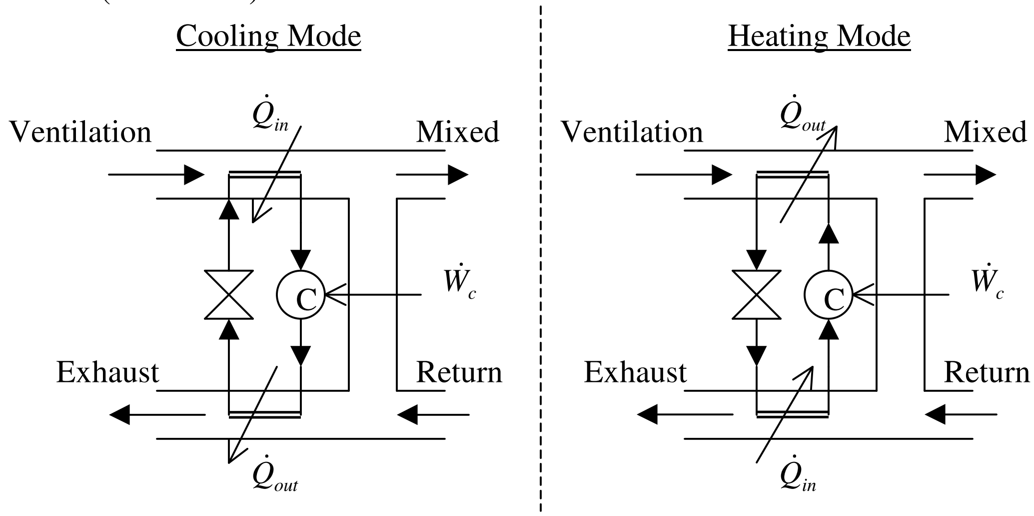


Figure 3. Heat Pump Heat Recovery (HPHR)

Literature Review

Emmerich (2001) performed an extensive literature review for DCV that is a valuable resource in understanding the development and application of DCV technology. With respect to evaluation of energy savings associated with DCV, there have been a number of simulation and field studies. Simulation studies were performed by Knoespel et al. (1991), Haghighat et al. (1993), Carpenter (1996), and Brandemuehl and Braun (1999). These studies demonstrated significant savings associated with the implementation of DCV for both small and large commercial buildings. The largest savings occur for buildings with highly variable occupancy, such as auditoriums and in more extreme climates (hot or cold) where ventilation loads are a larger fraction of the total loads. It is extremely important to use an economizer in conjunction with DCV so as not to lose any free cooling poetenail. For commercial buildings, the percent savings are greater for DCV during the heating season than the cooling season. Also, relative savings are greater for VAV systems than for CAV systems.

Field studies for DCV have been performed by Janssen et al. (1982), Gabel et al. (1986), Donnini et al. (1991), and Zamboni et al. (1991). The savings determined from field results have generally been consistant with the simulation results. The energy savings are significant and greater savings occur for buildings with highly variable occupancies, such as auditoriums. In some cases, the maximum occupancy was a small percentage of the design occupancy used to determine the fixed ventilation rates and the zone CO_2 concentrations never reached the set point. In these situations, infiltration and air leakage through the damper were sufficient to satisfy the ventilation requirements. In some cases, there were some occupant complaints of increased odor during DCV control.

Enthalpy exchangers were initially developed for commercial HVAC applications in the late 1970s. However, assessment of this technology has only recently appeared within the literature. Stiesch et al. (1995), Rengarajan et al. (1996), and Shirey et al. (1996)

evaluated enthalpy exchangers through simulation and found the technology to be economically viable. Greater potential was found for cooling in warm and humid climates. One of the significant factors affecting performance is pressure drop associated with air flow through the media. The additional fan power associated with application of this technology is significant.

Very few studies have been performed to evaluate the application of heat pumps for heat recovery in ventilation systems. Fehrm et al. (2002) estimated that for residential systems in Sweden and Germany, the use of heat pump heat recovery in a forced ventilation system would reduce energy consumption and peak demand by about 20% when compared to a conventional gas-fired boiler system.

Objective

Although individual case studies have been performed for DCV, enthalpy exchanger and heat pump heat recovery systems, the overall economics of these technologies have not been fully evaluated and compared. These are competing technologies and would not be implemented together. The overall objective of the work described in this report is to provide an economic assessment of these alternative ventilation strategies for a range of small commercial buildings in the state of California.

Assessment Approach

The primary approach for assessing the ventilation strategies was to perform detailed simulations to estimate operating costs, economic payback periods and rate of return. A simulation tool, termed the **V**entilation **S**trategy **A**ssessment **T**ool (VSAT) was developed to estimate cost savings associated with different ventilation strategies for small commercial buildings. A set of prototypical buildings and equipment is also part of the model. The tool is not meant for design or retrofit analysis of a specific building, but to provide a quick assessment of alternative ventilation technologies for common building types and specific locations with minimal user input requirements. The goal in developing VSAT was to have a fast, robust simulation tool for comparison of ventilation options that could consider large parametric studies involving different systems and locations. Existing commercial simulation tools do not consider all of the ventilation options of interest for this project.

The buildings considered within VSAT include a small office building, sit-down restaurant, retail store, school class wing, school auditorium, school gymnasium, and school library. All of these buildings are considered to be single zone with a slab on grade (no basement or crawl space). VSAT considers only packaged HVAC equipment, such as rooftop air conditioners with integrated cooling equipment, heating equipment, supply fan, and ventilation. Modifications to the ventilation system are the focus of the tool's evaluation.

Field sites were also established for the DCV and heat pump heat recovery systems. The goals of the field testing were to verify savings and to identify practical problems associated with these technologies. Several field sites were established for DCV that would allow side-by-side testing for different building types in different climates. A single field site was established for the heat pump heat recovery unit in order to verify the performance of the unit.

II. SIMULATION DESCRIPTION

Braun and Mercer (2003a) provide a detailed description of the models employed within VSAT along with validation results. The tool is based upon a program developed by Brandemuehl and Braun (2002). Figure 4 shows an approximate flow diagram for the modeling approach. Given a physical building description, an occupancy schedule, and thermostat control strategy, the building model provides hourly estimates of the sensible cooling and heating requirements needed to keep the zone temperatures at cooling and heating set points. This involves calculation of transient heat transfer from the building structure and internal sources (e.g., lights, people, and equipment). The air distribution model solves energy and mass balances for the zone and air distribution system and determines mixed air conditions supplied to the equipment. The mixed air condition supplied to the primary HVAC equipment depends upon the ventilation strategy employed. The zone temperatures are outputs from the building model, whereas the zone and return air humidities and CO_2 concentrations are calculated by the air distribution model. The equipment model uses entering conditions and the sensible cooling requirement to determine the average supply air conditions. The entering and exit air conditions for the air distribution and equipment models are determined iteratively at each timestep of the simulation using a non-linear equation solver. The economic model predicts hourly operating cost for each system employing a different ventilation strategy based on electrical and gas rate structures. Payback is calculated from annual results with respect to the base case strategy.

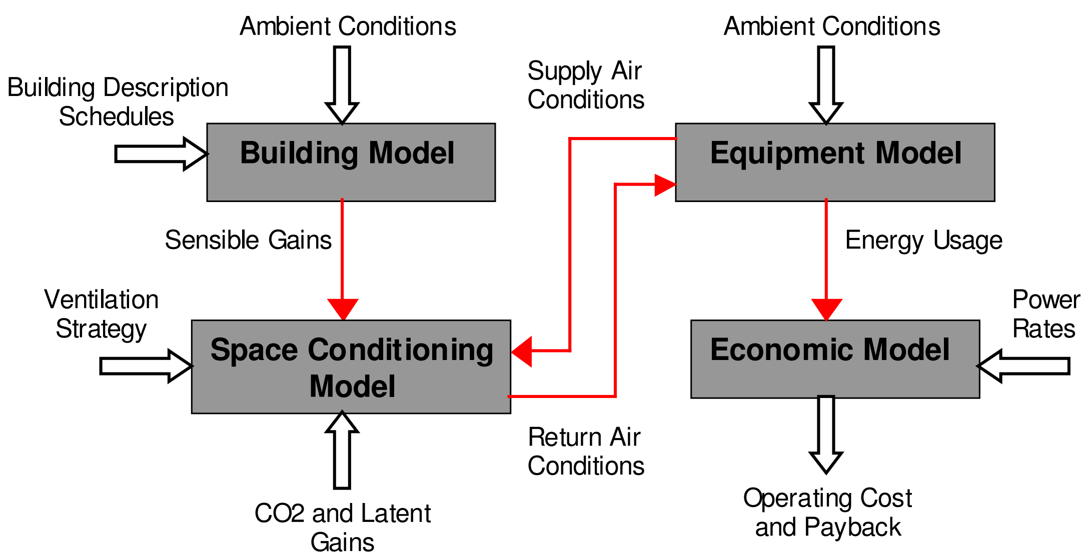


Figure 4. Diagram of VSAT Modeling Approach

Component Modeling Approaches

The building model involves detailed calculations that consider transient conduction through walls using transfer function representations. Predictions of the model compare well with other detailed models from the literature with substantially faster calculation speeds (Braun and Mercer, 2003a).

The space conditioning model follows the approach employed by Brandemuehl and Braun (1999) and employs the use of quasi-steady-state mass and energy balances on the air within the zone and air distribution system. A fixed ventilation effectiveness is employed for the zone to consider short-circuiting of supply air to the return duct. The DCV control is assumed to be ideal: the model determines the minimum ventilation air necessary to maintain the CO_2 set point. The base case and DCV systems employ a differential enthalpy economizer.

Both the primary air conditioning and heat pump heat recovery units are modeled using an approach similar to that incorporated in ASHRAE's *HVAC Toolkit* (Brandemuehl et al., 2000). The model for the primary air conditioner utilizes prototypical performance characteristics, which are scaled according to the capacity requirements and efficiency at design conditions. The characteristics of the heat pump heat recovery unit are based upon measurements obtained from the manufacturer and from tests conducted at the Herrick Labs, which are also scaled for different applications. Braun and Mercer (2002) describe the laboratory testing and development of the heat pump model.

The ventilation heat pump heat recovery unit is only enabled during occupied hours. During unoccupied hours, the primary air conditioner and heater must meet the cooling and heating requirements. In addition, the heat pump will only operate in cooling mode when the ambient temperature is above 68 F. When the heat pump is enabled, it provides the 1st stage for cooling or heating with the 2nd stage provided by the primary air conditioner or heater.

The enthalpy exchanger is modeled using an approach developed by Stiesch et al. (1995). This component model predicts temperature, humidity and enthalpy effectiveness based on a dimensionless wheel speed and media NTU. The enthalpy exchanger operates when the primary fan is on and the ambient temperature is less than 55 F or greater than the return air temperature. When the ambient temperature is between 55 F and the return air temperature, it is assumed that a cooling requirement exists and it is better to bring in cooler ambient air. When the ambient temperature is below 55 F, then a feedback controller adjusts the speed to maintain a ventilation supply air temperature of 55 F. When the ambient temperature is above the return air temperature, then the wheel operates at maximum speed. Feedback control of wheel speed is also initiated under conditions where water vapor in the exhaust stream would condense and freeze. A frost set point is specified based on winter ambient and zone design conditions as discussed by Stiesch (1995).

The primary supply fan operates at a fixed speed and is modeled assuming a constant fan/motor efficiency and overall pressure loss. An additional exhaust fan is included for systems utilizing a heat pump heat recovery unit or enthalpy exchanger.

VSAT was validated by comparing annual equipment loads and power consumptions for similar case studies in Energy-10 (Balcomb, 2002) and TRNSYS (2002). Energy-10 is a design tool developed for the U.S. Department of Energy (DOE) to analyze residential and small commercial buildings. TRNSYS is a complex transient system simulation program that incorporates a detailed building load model (Type-56 multi-zone building component). Neither of these tools incorporates the ventilation strategies considered in this study. Therefore, VSAT was validated for a base case employing the conventional ventilation strategies. In general, the VSAT predictions were within about

5% of the hourly, monthly, and annual predictions from TRNSYS and Energy-10 (Braun and Mercer, 2003a).

Modeling Parameters

The default parameters in VSAT were employed for the simulation results (medium efficiency equipment index with rated air conditioner EER of 9.5 and gas furnace efficiency of 0.75, supply fan power of 0.4 W/cfm, ventilation effectiveness of 0.85, and 350 ppm ambient air CO_2 concentration). The DCV system utilizes a set point for CO_2 concentration in the zone of 1000 ppm. With an 85% ventilation effectiveness, this leads to a return air CO_2 set point of approximately 900 ppm. The exhaust fan power for the enthalpy exchanger and heat pump is 0.5 W/cfm for each unit. Appendix A contains detailed descriptions of the prototypical buildings that are employed within VSAT.

Weather Data

VSAT includes weather data for the California climate zones shown in Figure 5. The representative cities for each climate zone (CZ) are given in Table 1. The climate zones are based on energy use, temperature, weather and other factors. They are basically a geographic area that has similar climatic characteristics. The California Energy Commission (CEC) originally developed weather data for each climate zone by using unmodified (but error-screened) data for a representative city and weather year (representative months from various years). The CEC analyzed weather data from weather stations selected for (1) reliability of data, (2) currency of data, (3) proximity to population centers, and (4) non-duplication of stations within a climate zone. There are two sets of climate zone data included in VSAT, the original and a massaged set. In the massage data, the dry bulb temperature has been modified in an effort to give the file a better "average" across the entire zone. However, because only dry bulb was adjusted, the humidity conditions are affected and therefore, the massaged files are not preferred. The original data set was used for the results presented in this report.

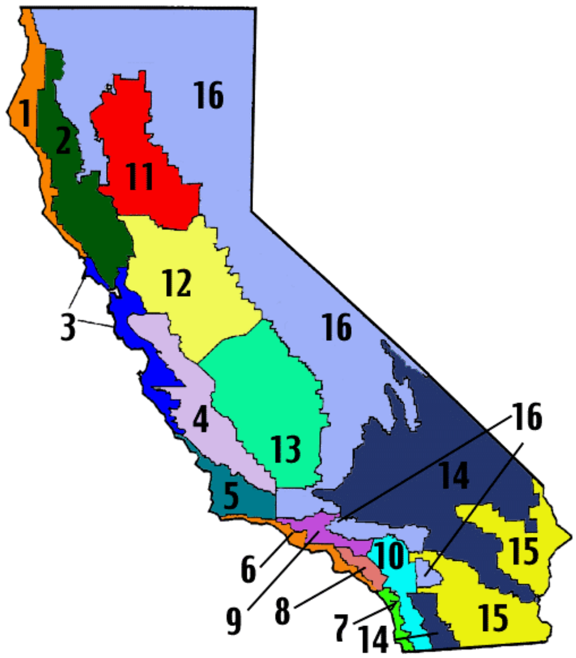


Figure 5. California Climate Zones

Table 1. Cities Associated with California Climate Zones

CZ 1: Arcata	CZ 5: Santa Maria	CZ 9: Pasadena	CZ13: Fresno
CZ 2: Santa Rosa	CZ 6: Los Angeles	CZ10: Riverside	CZ14: China Lake
CZ 3: Oakland	CZ 7: San Diego	CZ11: Red Bluff	CZ15: El Centro
CZ 4: Sunnyvale	CZ 8: El Toro	CZ12: Sacramento	CZ16: Mount Shasta

Economic Analysis

Operating costs associated with each ventilation strategy are calculated based on annual electric power and/or gas consumption by the HVAC equipment. Percent savings for each ventilation strategy are assessed by comparing annual operating costs to the base case. For retrofit applications, simple payback period is used to compare technologies. However, for new buildings, a cumulative rate of return is the performance indice used for comparisons.

The annual operating costs for an HVAC system within VSAT are calculated assuming a three tiered utility rate structure of on-peak, mid-peak and off-peak rates. These costs are calculated according to

$$C_k = \sum_{m=1}^{m=12} \left\{ \begin{array}{c} r_{d,on,m} \cdot \dot{W}_{peak,on,m} + r_{d,mid,m} \cdot \dot{W}_{peak,mid,m} + r_{d,off,m} \cdot \dot{W}_{peak,off,m} \\ + \sum_{i=1}^{N_m} \left(r_{e,i,m} \cdot W_{i,m} + r_{g,i,m} \cdot G_{i,m} \right) \end{array} \right\} \tag{1}$$

where subscript k denotes the HVAC system associated with a particular ventilation strategy k, m is the month and i is the hour of the year. N_m is the number of hours within month m. For each month m, $r_{d,on,m}$, $r_{d,mid,m}$ and $r_{d,off,m}$ correspond to the utility rates for electricity demand during the on-peak, mid-peak and off-peak time periods ($/kW). Peak power consumption for the HVAC equipment during the on-peak, mid-peak and off-peak periods is represented as $\dot{W}_{peak,on,m}$, $\dot{W}_{peak,mid,m}$ and $\dot{W}_{peak,off,m}$, respectively. For each hour *i* of month *m*, r_e is the utility rate associated with electricity usage ($/kWh), *W* corresponds to the amount of electricity consumed (kWh), r_g is the utility rate associated with natural gas usage ($/therm) and *G* represents the amount of gas consumed (therm).

Annual electricity costs include both energy ($/kWh) and demand charges ($/kW). Gas energy usage costs do not vary with time of the day. However, the user may enter different electric and gas rates for summer and winter periods. The user may also adjust the start month for the summer and winter periods and the times of day associated with on-peak, mid-peak and off-peak periods.

Each ventilation strategy is compared with an assumed base case of fixed ventilation incorporating a setup/ setback thermostat and differential enthalpy economizer. Annual operating cost savings (S_k) for each ventilation strategy *k*, when compared to the base case, are calculated according to

$$S_k = C_{BASE.CASE} - C_k \tag{2}$$

Annual operating cost percent savings ($\%S_k$) for each ventilation strategy *k* are calculated according to

$$\%S_k = \left\{ 1 - \frac{C_k}{C_{BASE.CASE}} \right\} \cdot 100\% \tag{3}$$

For retrofit analysis, the economics of the different technologies only depend upon the initial costs of the equipment and the energy cost savings. In this case, simple yearly payback (N_{pb}) for each ventilation strategy *k* is calculated according to

$$N_{pb} = \frac{I_k}{S_k} \tag{4}$$

where I_k is the first cost, including installation and any equipment, associated with implementing ventilation strategy *k*. If annual operating cost savings for any ventilation strategy are negative, implying the base case is less expensive to operate, payback is not calculated.

For new buildings, additional cost savings can be realized for the enthalpy exchanger and heat pump through reductions in the size of the primary heating and cooling equipment. In this case, payback periods are not a very good performance indice for comparison and rate of return was employed instead. The cumulative rate of return (RR_k) for each ventilation strategy *k* is calculated according to

$$RR_k = \left(\frac{QS_k - I_k + (N_{cum} \cdot S_k)}{I_k} \right) \cdot 100\% \tag{5}$$

where QS_k is the savings in equipment cost ($) due to primary RTU downsizing compared the base case and N_{cum} represents the number of years (cumulative years) used in calculating the rate of return. The HXHR and HPHR systems require smaller primary RTUs because of energy recovery in the ventilation streams. However, the DCV requires the same equipment capacity as the base case (QS_k =$0) because the system must be able to handle the design ventilation requirement at design conditions.

All utility rates used for economic results assume secondary, firm service (electricity constantly supplied) and a monthly electric demand less than 500 kW. Typical utility rate information was obtained for small commercial service in each of the California climate zones and implemented within VSAT. Table 2 summarizes the utility rates that were considered for each climate zone. Pacific Gas and Electricity (PGE), Southern California Edison (SCE), Southern California Gas (SCG) and San Diego Gas and Electricity (SDGE) are the major utility suppliers in California. The utility rates of each supplier differ depending upon time-of-use. Table 3 shows the time-of-use associated with each utility provider. The cities associated with climate zones 10 (Riverside) and 15 (El Centro) are served by local energy companies. However, for the electric rate structure within VSAT, Southern California Edison was assumed for both climate zones 10 and 15 because the majority of CZ 10 and approximately half of CZ 15 is territory within the service area of Southern California Edison. Southern California Gas Company was also assumed for most all the southern California climate zones except CZ 07, which is serviced by San Diego Gas and Electricity.

For summer electricity consumption, the demand charge for Pacific Gas and Electricity is higher, almost twice that of Southern California Edison; while Pacific Gas and Electricity's energy charge is low, only half of Southern California Edison's energy charge. For Pacific Gas and Electric, the ratio of on-peak to off-peak demand charges is greater than 5, whereas Southern California Edison does not charge demand fees during off-peak times. For energy charges, both companies have on-peak to off-peak ratios of about 2. San Diego's time-of-use energy charge ratio is much lower.

Table 2. Utility Rates in California

CZ	Representative City	Service Provider	Time of Use	Summer Season	Winter Season
1	Arcata		*Demand Charge- $/kW*		
2	Santa Rosa		On Peak	$13.35	N/A
3	Oakland	Pacific Gas	Mid Peak	$3.70	$3.65
4	Sunnyvale	And	Off Peak	$2.55	$2.55
5	Santa Maria	Electricity	*Energy Charge - $/kWh*		
11	Red Bluff	(Schedules E-19	On Peak	0.0877	N/A
12	Sacramento	and G-NR1)	Mid Peak	0.0581	0.0639
13	Fresno		Off Peak	0.0506	0.0504
			Gas Charge - $/therm		
				$0.6736	$0.7422
6	Los Angeles		*Demand Charge- $/kW*		
8	El Toro	Southern	On Peak	$7.75	$0.00
9	Pasadena	California Edison	Mid Peak	$2.45	$0.00
10	Riverside	(Schedule TOU-	Off Peak	$0.00	$0.00
14	China Lake	GS-2) and	*Energy Charge - $/kWh*		
15	El Centro	Southern	On Peak	0.2960	N/A
16	Mount Shasta	California Gas	Mid Peak	0.1176	0.1296
		(Schedule GN-10)	Off Peak	0.0942	0.0942
			Gas Charge - $/therm		
				$0.7079	$0.7079
7	San Diego		*Demand Charge- $/kW*		
			On Peak	$10.42	$4.83
		San Diego Gas	Mid Peak	N/A	N/A
		and Electricity	Off Peak	N/A	N/A
		(Schedules AL-	*Energy Charge - $/kWh*		
		TOU and EECC	On Peak	$0.1163	$0.1151
		and GN-3)	Mid Peak	$0.0895	$0.0894
			Off Peak	$0.0884	$0.0884
			Gas Charge - $/therm		
				$0.6524	$0.7497

Table 3. Time-Of-Use for California Utility Companies

PGE			
Summer:	May 1 - Oct. 31	Winter:	Nov. 1 - April 30
On-Peak	12:00 - 6:00, M - F	On-Peak	N/A
Mid-Peak	8:00 AM - 12:00 &	Mid-Peak	8:00 AM - 9:00 PM, M - F
	6:00 PM - 9:00 PM, M - F		
Off-Peak	9:00 PM - 8:00 AM, all week	Off-Peak	9:00 PM - 8:00 AM, all week
SCE			
Summer:	June 1 - Sept. 30	Winter:	Oct. 1 - May 31
On-Peak	12:00 - 6:00, M - F	On-Peak	N/A
Mid-Peak	8:00 AM - 12:00 &	Mid-Peak	8:00 AM - 9:00 PM, M - F
	6:00 PM - 11:00 PM, M - F		
Off-Peak	11:00 PM - 8:00 AM, all week	Off-Peak	9:00 PM - 8:00 AM, all week
SDGE - Electric Rate			
Summer:	May 1 - Sept. 30	Winter:	Oct. 1 - April 30
On-Peak	11:00 - 6:00, M - F	On-Peak	5:00 - 8:00, M - F
Mid-Peak	6:00 AM - 11:00 &	Mid-Peak	6:00 AM - 5:00 PM &
	6:00 PM - 10:00 PM, M - F		8:00 PM - 10:00 PM, M - F
Off-Peak	10:00 PM - 6:00 AM, all week	Off-Peak	10:00 PM - 6:00 AM, all week
SDGE - Gas Rate			
Summer:	April 1 - Nov. 30	Winter:	Dec. 1 - March 31
SCG			
Summer:	April 1 - Nov. 30	Winter:	Dec. 1 - March 31

First costs for demand controlled ventilation, the heat pump and enthalpy exchanger were obtained from personal contact with representatives from each specific equipment manufacturer. The first costs included equipment and installation costs associated with each ventilation strategy.

For DCV, the number of rooftop units employed for a particular HVAC system must be known in order to determine the associated first costs. It is difficult to ascertain how many DCV controllers, or rooftop units are necessary for a given application. This situation is very sensitive to the dynamics of the duct runs, availability of space and actual number of RTUs that may or may not accommodate the specific building. The economic analysis assumed that RTUs are available in sizes of 5, 7.5, 10, 15 and 20 ton cooling capacities. For a simulated prototypical building and location, the number of individual RTUs was determined based upon utilizing the fewest possible number of units necessary to realize a cooling capacity that was within a target range of 5% of the sized equipment cooling capacity. First costs for DCV included a DCV logic controller, zone CO_2 sensor and 4 hours time for installation. These costs combined were estimated at $900 per RTU within VSAT.

The heat pump first costs were based on the actual equipment, installation, controls and thermostat costs. Based on correspondence with manufacturer's representatives, $5 per/cfm ventilation air was assumed for calculating a generalized first cost of the heat pump.

The enthalpy exchanger first costs include the same elements as for the heat pump, however, $2 per/cfm ventilation air is assumed. Enthalpy exchangers do not require as many components as a heat pump and are easier to manufacture, therefore the equipment cost is lower.

A value of $1000 per ton was assumed for installed costs of RTUs in calculating the equipment cost savings for new building applications. Savings in primary heater costs were not considered.

III. SIMULATION RESULTS

Sample Hourly Results

Systems with DCV generally have higher zone CO_2 concentrations because of lower ventilation rates. Figure 6 and Figure 7 show example weekday hourly CO_2 levels for the different ventilation strategies when applied to the restaurant and office building prototypes. These example days were simulated on July 19 in CZ 15. The zone CO_2 levels track the occupancy schedules and are identical for the base case, HPHR and HXHR cases because the ventilation rates are identical. The CO_2 levels for the base case can be lower than the HXHR and HPHR levels when the economizer operates, however, economizer operation did not occur on this particular day. The CO_2 levels are higher for the DCV+EC strategy due to lower ventilation rates. For the restaurant, the CO_2 concentrations were at the set point for a large portion of the occupied period. However, the set point was not reached for the office on this day. For the office, the average occupancy was low enough that infiltration (0.05 cfm/ft^2) provided sufficient fresh air to keep CO_2 levels in the zone below 1000 ppm. If infiltration did not exist, then an outdoor air fraction of about 0.06 would be necessary, on average, during the occupied period to maintain the zone CO_2 concentration at 1000 ppm in the office.

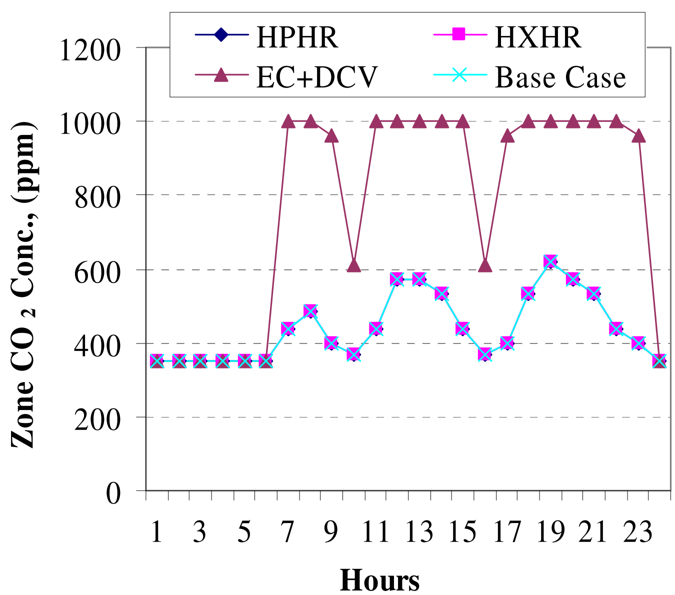


Figure 6. Hourly Zone CO_2 Concentration – Restaurant

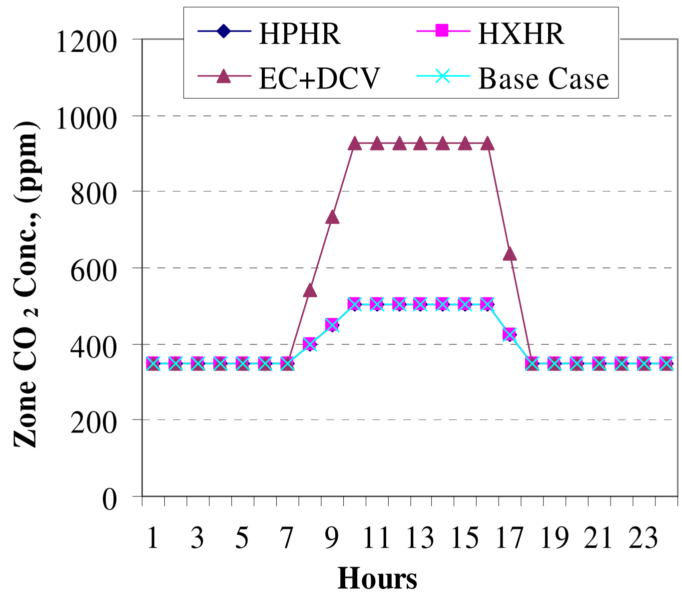


Figure 7. Hourly Zone CO_2 Concentration - Office

The different systems also lead to different humidity levels in the zone. Both the heat pump and enthalpy exchanger can remove moisture from the ventilation stream during the cooling season. The use of DCV can also lead to lower moisture levels when the ambient air is more humid than the zone air. Figure 8 and Figure 9 show sample hourly relative humidities for the restaurant in CZ 06 and the office in CZ 15 during summer.

For the more humid CZ 06, DCV plus economizer gave the lowest humidity levels except during economizer operation (e.g., morning hours for the restaurant). Also, the enthalpy exchanger had greater moisture removal from the ventilation air than the heat pump during occupied mode for CZ 06.

The heat pump and base case gave the lowest humidity levels in CZ 15 because this is a dry climate and the ventilation did not introduce an additional latent load. The heat pump did not dehumidify the air on this day and therefore the relative humidity in the zone was the same for the HPHR and base case case systems. The enthalpy exchanger actually transferred moisture from the exhaust stream and humidified the ventilation air. Thus, zone relative humidities for the enthalpy exchanger were higher than the base case for CZ 15. DCV+EC leads to lower outside air and therefore humidity levels in the zone were higher than for to the other ventilation strategies in this dry climate.

Clearly, the impact of the ventilation technology on zone humidity levels is very dependent on the climate. Both DCV and the HXHR systems provide higher humidity levels in dry climates and lower humidity levels in more humd climates than the base case. Both of these trends are good. The HPHR system provides lower humidity levels in humid climates and the same humidity levels in dry climates as the base case.

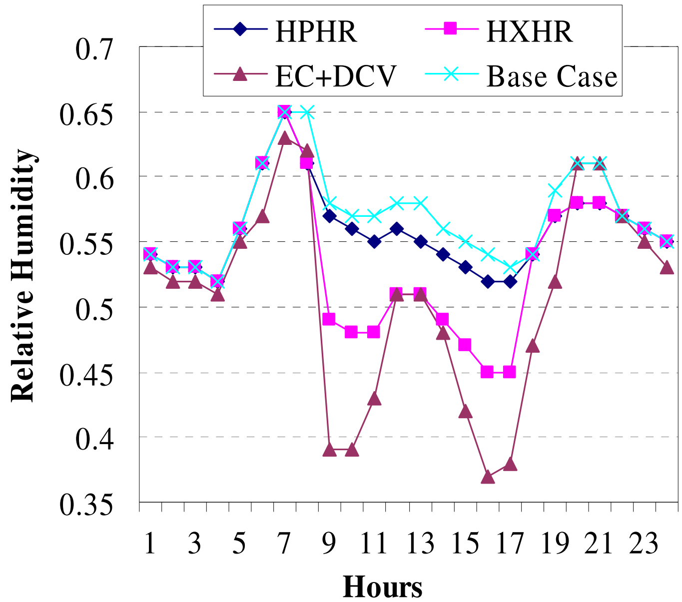


Figure 8. Relative Humidity – Restaurant, CZ06, June 20

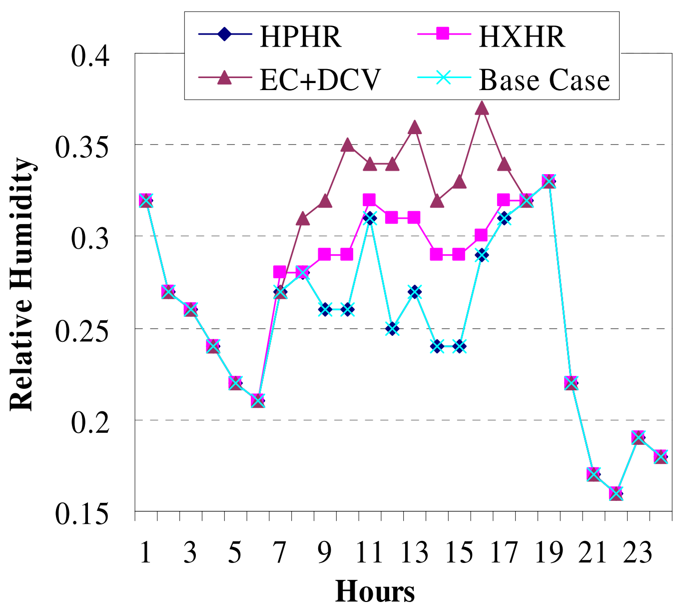


Figure 9. Relative Humidity – Office, CZ15, June 20

Implementation of any particular ventilation strategy should reduce the load and power consumption associated with the primary air conditioner. However, for the HPHR system, this power reduction is at the expense of power usage associated with the HPHR compressor. For both the HPHR and HXHR systems, an additional exhaust fan is also required to provide proper exhaust air flowrates for heat and mass exchange. The wheel medium and extra heat exchanger typically add 0.5 to 0.9 inches H_2O of pressure drop that must be overcome. The total fan power consumption of the heat pump or enthalpy exchanger plays a significant role in determining if either of these ventilation strategies is competitive when compared to the base case.

Figure 10 shows an example of hourly power consumption for the restaurant. The fan power for the heat pump and enthalpy exchanger is notably increased over the fan power for the base case and DCV+EC strategies. The "compressor power" includes power usage for the primary AC compressor and condenser fan and the HPHR compressor (for HPHR case). Although the total AC and heat pump compressor power input is slightly less than the compressor power for the base case, it is not sufficient to offset the increase in fan power at any time of the day for this example. However, for the HXHR system, the decrease in AC compressor power does overcome the additional power required for the exhaust fan.

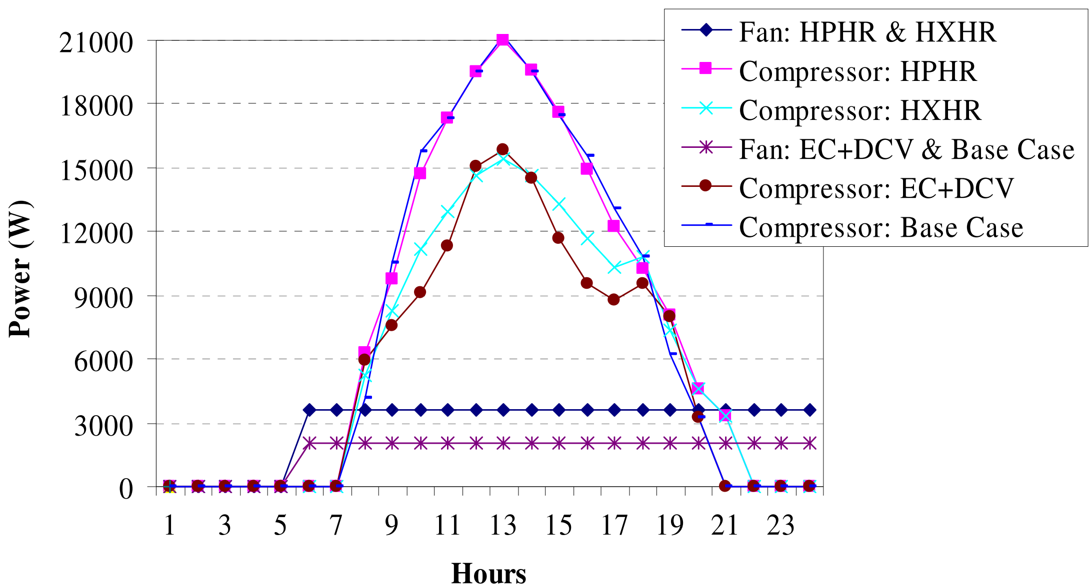


Figure 10. Equipment Power Consumption – Restaurant, CZ06, June 20

Each ventilation strategy also reduces primary energy consumption associated with heating. However, for the HPHR and HXHR systems, these reductions are offset by increases in electric power consumption. Figure 11 shows example hourly gas consumption for each of the strategies for the restaurant on January 20 in CZ 16. Figure 12 shows the corresponding electrical power consumption associated with each strategy and the base case for the same day. For this example day, all of the strategies result in reduced gas consumption when compared with the base case. However, the DCV+EC strategy results in the lowest gas consumption and there is no penalty associated with increased power requirements. From Figure 12, the power for the HPHR system is considerably higher than the power for the base case due to the additional compressor and fan. The power for the HXHR system is also greater than the base case because of the additional fan requirement.

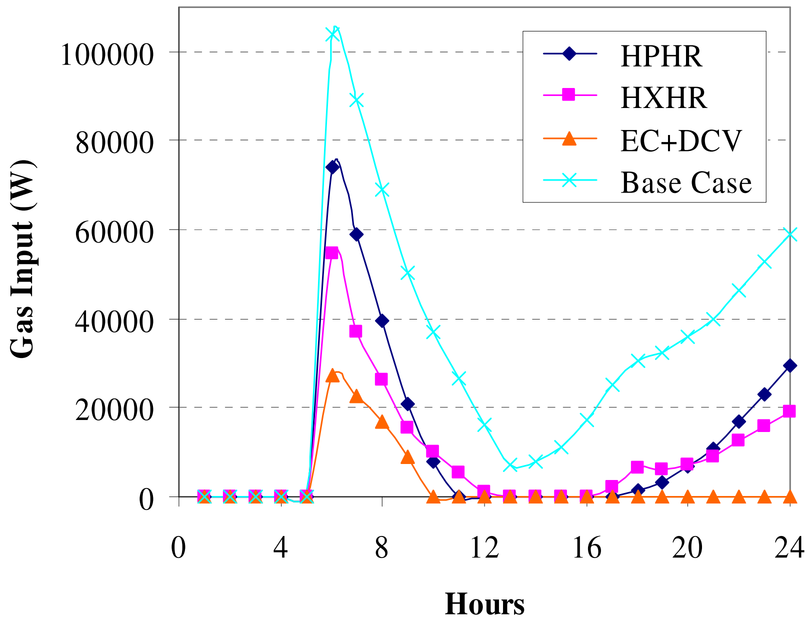


Figure 11. Furnace Gas Input – Restaurant, CZ16, Jan. 20

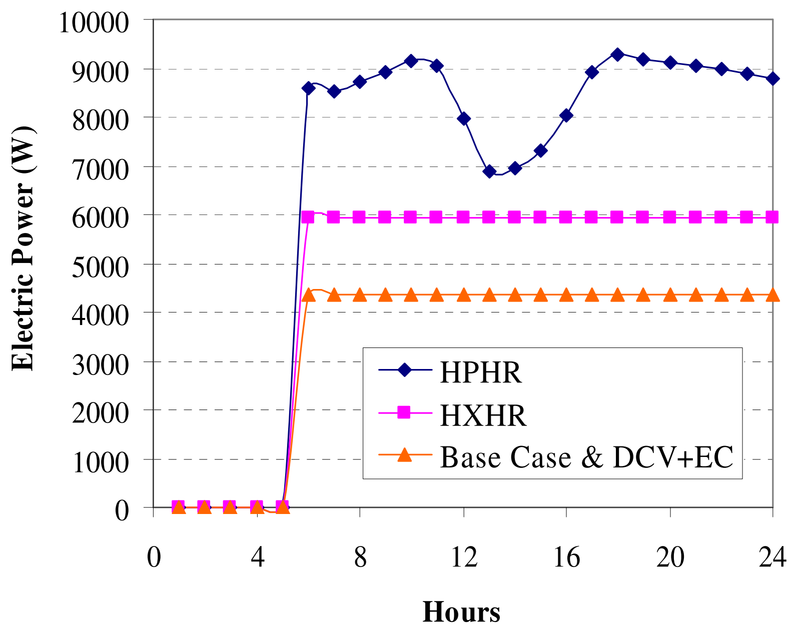


Figure 12. Electric Power Input – Restaurant, CZ16, Jan. 20

Figure 13 shows the daily operating cost for this example day for each ventilation strategy. All of the strategies result in some overall savings for this day when compared to the base case. However, HPHR savings are very small. As ambient temperatures get colder and occupied periods last longer, the heat pump performs much better and approaches the performance of the enthalpy exchanger. CZ 16 requires the most heating when compared to all other climate zones within California. Since California is a mild climate, the savings potential of the HPHR technology is not very significant when compared to the savings potential in other colder areas of the United States. This consequence will be further investigated in a later section.

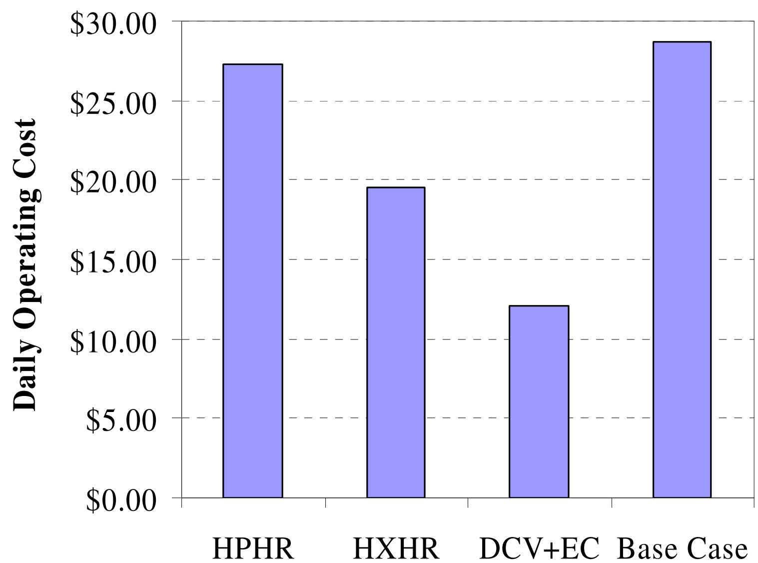


Figure 13. Daily Restaurant Operating Costs, CZ16, Jan. 20

Annual Operating Cost Savings

The cost savings associated with demand-controlled ventilation and an economizer (DCV+EC), heat pump heat recovery (HPHR) and the enthalpy exchanger (HXHR) were compared on a percent savings basis relative to the assumed base case (fixed ventilation with a differential economizer). Appendix B gives annual energy usage and costs for the base case applied to the seven prototypical buildings in the 16 California climate zones. The percent savings were calculated according to

$$Y = \left[1 - \frac{X.vent.strategy}{X.base.case}\right] * 100\% \tag{5}$$

where; Y = relative percent savings

$X.vent.strategy$ = quantity under consideration for the specific ventilation strategy (DCV+EC, HXHR, or HPHR)

$X.base.case$ = quantity under consideration for the base case

Table 4 through Table 10 give percent savings for each of the strategies applied to all prototypical buildings in all climate zones assuming a retrofit application. Four quantities are compared for each building type: total electrical energy costs, electric demand costs, gas costs and total equipment operating costs. Negative savings imply that the strategy had greater costs than the base case.

The greatest savings potential for all the building types is associated with demand-controlled ventilation with an economizer. For DCV+EC, the ventilation load is directly related to the occupancy schedule. For most buildings, the average occupancies are much lower than the design occupancy used to determine fixed ventilation rates. The total cost savings for DCV+EC ranged from about 1% to 48%, whereas the electrical demand savings were between 1% and 52%. The greatest savings for DCV+EC among the building types occurred for the school auditorium and school gym. Both of these building types have intermittent occupancy schedules and average occupancies that are a small fraction of the peak design occupancy. The heating load for DCV is practically eliminated for several building types where internal gains tend to balance other heat losses from the building. Even greater overall DCV+EC cost savings would be expected in climates that have significantly greater heating loads than occur for California.

The enthalpy exchanger was the next most effective ventilation strategy for the cases considered. The percent savings in gas costs are significant in most cases. However, gas costs are relatively low for these climates and therefore these savings have a relatively small impact on total savings. In most cases, the electrical energy costs are higher for the HXHR system than for the base case due to two effects: 1) increased fan energy and 2) loss of significant free cooling opportunities without an economizer. However, there are significant demand cost savings in many cases. The greatest electrical energy and demand cost savings occur for buildings and locations that have the highest ventilation loads. Positive total cost savings occurred for the central and eastern portions of the state for the restaurant, retail store, auditorium and gym. These regions have the most extreme ambient temperatures and these buildings have the highest peak occupant densities. With high ambient temperatures, there is less opportunity for economizer operation and better opportunities for heat recovery. Both of these effects tend to increase savings associated with the HXHR system.

The trends for the heat pump heat recovery system are similar to the HXHR system, but the overall performance is worse. The savings in gas consumption are actually greater than those for the HXHR, but at the expense of increased electrical usage for heating. In almost every situation, the HPHR system had greater overall operating costs than the base case. In general, the cooling COP for the HPHR unit increases with ambient wet bulb temperature whereas the heating COP increases with decreasing ambient temperature. The performance of the HPHR unit needs to be "good enough" so that primary equipment savings offset increases in electrical energy due to the HPHR compressor and exhaust fan. Overall cooling savings only occur at very high ambient wet bulb temperatures. For heating, positive savings can be at relatively moderate ambient temperatures. However, the California climate zones are all relatively moderate and any savings associated with heating are not sufficient to offset increases in cooling season costs.

Table 4. Office Savings Comparisons

	Demand Controlled Ventilation + EC				**Heat Pump Heat Recovery**				**Enthalpy Exchanger**			
	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %
CACZ01	-0.9	7.6	54.1	9.2	-66.7	-11.1	29.8	-22.5	-65.0	-8.4	20.0	-21.2
CACZ02	3.6	6.6	50.0	7.7	-20.6	-3.3	29.8	-7.8	-18.1	1.1	24.8	-4.5
CACZ03	2.3	8.4	56.8	7.5	-37.0	-5.7	36.4	-15.5	-35.7	1.6	22.7	-10.8
CACZ04	8.0	13.7	53.6	12.5	-20.0	-2.7	34.4	-8.4	-15.9	6.1	25.6	-1.9
CACZ05	0.4	2.6	50.0	2.7	-31.6	-2.0	36.5	-12.1	-30.2	2.7	24.3	-8.9
CACZ06	5.0	13.8	66.7	6.6	-23.5	-2.8	50.0	-20.1	-21.9	6.3	25.0	-17.4
CACZ07	5.6	6.1	66.7	5.9	-24.9	-10.4	55.6	-20.6	-22.4	-2.9	33.3	-16.7
CACZ08	6.9	14.6	51.6	8.3	-16.2	-1.6	38.7	-13.6	-12.3	8.1	22.6	-9.0
CACZ09	7.1	15.5	70.0	8.6	-14.9	0.6	55.0	-12.2	-10.9	9.3	40.0	-7.5
CACZ10	6.9	11.0	45.2	7.9	-12.4	0.9	30.6	-10.0	-9.0	6.2	21.0	-6.4
CACZ11	3.8	2.3	50.0	5.0	-14.9	-2.0	29.4	-5.7	-12.2	1.3	24.4	-3.0
CACZ12	6.0	10.0	51.2	10.0	-16.7	-2.5	27.7	-6.9	-13.3	2.3	23.9	-3.0
CACZ13	6.7	9.5	51.8	9.4	-11.3	-1.4	28.9	-4.8	-7.8	3.7	24.7	-0.6
CACZ14	3.0	7.4	43.5	5.1	-9.8	1.3	23.1	-6.9	-7.8	5.4	25.4	-4.6
CACZ15	8.4	11.8	54.5	9.0	-4.3	1.2	40.9	-3.5	-0.3	7.3	31.8	0.7
CACZ16	3.7	7.2	44.4	10.8	-22.6	-0.9	23.5	-11.7	-18.2	5.1	24.9	-7.6

Table 5. Restaurant Savings Comparisons

	Demand Controlled Ventilation + EC				Heat Pump Heat Recovery				Enthalpy Exchanger			
	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %
CACZ01	-21.3	1.4	99.9	35.0	-139.1	-19.8	90.8	-0.8	-89.4	-13.3	43.7	-6.4
CACZ02	3.4	10.3	99.4	24.1	-41.6	-2.8	83.3	-0.8	-26.4	8.4	56.2	4.8
CACZ03	-5.3	9.2	100.0	17.9	-64.3	-7.9	92.1	-11.3	-50.4	6.6	47.1	-5.7
CACZ04	5.9	21.1	99.9	23.4	-37.1	-1.7	86.1	-6.3	-24.4	16.6	56.6	4.9
CACZ05	-8.7	3.4	100.0	12.3	-58.7	-2.4	88.0	-9.3	-47.5	5.9	53.0	-5.9
CACZ06	0.0	20.6	100.0	8.0	-42.7	-6.2	94.8	-30.4	-36.9	13.2	42.4	-25.4
CACZ07	2.6	15.2	100.0	10.2	-44.9	-12.7	93.3	-30.3	-37.1	4.3	51.3	-22.3
CACZ08	6.9	21.4	100.0	12.8	-31.0	-1.9	91.0	-21.8	-21.8	17.4	54.5	-13.0
CACZ09	9.2	22.1	100.0	13.8	-25.1	0.1	92.0	-17.9	-16.0	19.0	58.7	-8.7
CACZ10	10.6	22.2	100.0	16.3	-20.2	4.2	86.5	-12.1	-11.0	18.6	61.1	-3.8
CACZ11	8.1	11.7	99.3	22.7	-27.6	2.0	84.4	1.3	-14.9	12.8	57.6	7.6
CACZ12	8.1	15.3	99.7	24.2	-31.6	-2.2	85.7	-1.8	-18.3	11.4	56.5	5.7
CACZ13	12.2	18.6	99.8	22.9	-20.1	0.4	84.9	-1.4	-8.8	14.0	58.8	7.6
CACZ14	9.6	16.5	98.4	20.1	-18.4	7.8	74.1	-5.3	-6.7	19.5	64.3	4.1
CACZ15	18.4	25.4	100.0	20.2	-5.3	5.7	87.5	-2.9	5.4	22.2	65.9	8.0
CACZ16	3.5	16.6	95.1	35.8	-54.4	2.5	70.9	-5.9	-28.7	14.3	62.3	6.7

Table 6. Retail Store Savings Comparisons

	Demand Controlled Ventilation + EC				**Heat Pump Heat Recovery**				**Enthalpy Exchanger**			
	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %
CACZ01	-17.8	11.5	100.0	23.4	-97.3	-19.3	88.2	-15.4	-76.3	-12.0	56.0	-13.3
CACZ02	6.9	18.7	100.0	22.2	-34.1	-2.5	80.0	-6.4	-23.8	9.9	66.3	2.6
CACZ03	-2.2	19.0	100.0	16.6	-53.3	-8.7	88.8	-18.3	-47.2	6.8	61.7	-8.8
CACZ04	9.0	31.8	100.0	25.7	-31.4	-2.6	82.0	-10.5	-22.0	16.2	69.2	3.1
CACZ05	-6.4	11.7	100.0	8.8	-48.4	-3.0	82.4	-16.2	-44.7	5.4	70.0	-10.5
CACZ06	1.4	31.0	100.0	6.7	-34.4	-6.6	93.0	-29.1	-31.7	12.4	73.6	-24.2
CACZ07	4.4	21.3	100.0	9.7	-38.5	-11.9	93.6	-30.4	-33.6	4.9	85.6	-22.4
CACZ08	8.8	33.4	100.0	13.4	-27.0	-2.3	86.7	-22.1	-19.5	17.7	74.6	-13.0
CACZ09	11.0	35.7	100.0	15.2	-23.7	0.7	93.2	-19.4	-15.2	19.6	85.9	-9.4
CACZ10	14.2	31.1	100.0	17.8	-17.4	3.1	81.4	-13.1	-9.0	17.8	74.9	-4.1
CACZ11	11.1	12.5	100.0	19.3	-23.2	1.7	82.5	-2.3	-12.6	12.9	65.5	6.3
CACZ12	11.3	23.9	100.0	24.2	-26.8	-1.0	82.9	-5.8	-16.5	12.2	66.1	4.0
CACZ13	16.3	28.1	100.0	26.0	-16.8	-0.2	81.7	-4.1	-6.6	13.9	67.7	6.9
CACZ14	12.4	18.1	99.7	18.6	-14.8	7.5	70.9	-6.7	-5.4	18.9	70.7	2.3
CACZ15	22.6	35.7	100.0	24.3	-4.8	5.3	85.4	-3.4	5.8	21.6	86.7	7.8
CACZ16	5.9	23.7	99.1	32.2	-46.1	0.4	70.6	-10.0	-26.8	13.6	65.5	2.2

Table 7. School Library Savings Comparisons

	Demand Controlled Ventilation + EC				**Heat Pump Heat Recovery**				**Enthalpy Exchanger**			
	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %
CACZ01	-5.8	4.3	75.8	17.1	-78.4	-15.7	63.0	-14.3	-63.7	-10.2	28.5	-14.8
CACZ02	3.1	6.5	74.0	11.4	-26.5	-2.9	55.3	-6.0	-19.9	4.1	36.0	-1.7
CACZ03	-1.0	9.3	82.9	10.3	-46.4	-5.9	62.9	-15.2	-41.9	4.9	31.4	-9.0
CACZ04	5.9	14.9	82.7	14.6	-25.5	-2.6	58.0	-8.7	-18.7	10.2	38.3	0.3
CACZ05	-3.3	-0.1	88.1	2.0	-38.8	-2.6	61.9	-13.3	-36.4	3.3	42.9	-9.6
CACZ06	1.3	17.2	100.0	4.4	-33.1	-2.1	75.0	-27.6	-30.8	10.5	50.0	-23.7
CACZ07	3.7	8.5	100.0	5.5	-34.0	-10.5	80.0	-26.8	-30.5	0.6	60.0	-21.2
CACZ08	5.5	16.6	94.1	8.0	-22.0	-2.3	64.7	-18.2	-16.6	11.3	47.1	-11.7
CACZ09	5.8	17.4	100.0	8.2	-20.2	0.3	76.9	-16.4	-14.4	12.2	53.8	-9.8
CACZ10	7.4	13.0	90.0	9.3	-15.1	0.3	56.7	-11.9	-9.9	9.0	46.7	-6.4
CACZ11	4.8	4.4	66.3	9.6	-18.1	-0.9	53.6	-3.4	-12.1	4.2	33.1	0.0
CACZ12	5.9	11.0	71.4	13.4	-22.1	-1.5	55.7	-5.5	-15.3	6.2	34.3	-0.3
CACZ13	8.0	10.6	73.2	12.6	-13.5	-1.6	56.3	-3.9	-7.0	7.0	34.8	2.3
CACZ14	4.3	8.4	70.3	8.6	-11.6	0.8	46.2	-6.5	-7.1	6.2	41.4	-2.5
CACZ15	10.0	13.7	88.9	10.7	-5.3	1.4	66.7	-4.3	0.5	8.9	55.6	1.7
CACZ16	3.5	11.7	56.1	17.7	-31.6	-1.6	41.9	-9.2	-20.3	6.6	35.5	-2.7

Table 8. School Gym Savings Comparisons

	Demand Controlled Ventilation + EC				**Heat Pump Heat Recovery**				**Enthalpy Exchanger**			
	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %
CACZ01	-11.5	1.7	59.8	24.8	-166.1	-38.3	66.8	-8.7	-89.3	-21.1	15.2	-13.7
CACZ02	8.3	13.6	59.2	19.4	-32.4	-5.7	58.0	-2.2	-16.6	6.4	23.2	3.6
CACZ03	2.3	16.8	66.7	21.2	-56.5	-11.0	69.7	-8.4	-38.0	5.1	17.6	-1.8
CACZ04	14.2	27.2	65.3	27.6	-29.2	-4.7	62.4	-4.4	-14.5	15.5	22.5	8.6
CACZ05	-0.4	12.2	73.0	14.8	-44.3	-4.9	68.2	-7.3	-33.4	4.3	19.9	-3.0
CACZ06	8.8	28.3	88.2	16.3	-29.6	-2.5	80.3	-19.2	-23.4	11.0	22.2	-13.6
CACZ07	11.8	26.3	94.4	19.4	-27.5	-4.3	88.7	-15.8	-19.1	16.0	31.5	-4.6
CACZ08	14.3	30.7	79.8	20.1	-18.4	-3.6	71.1	-12.3	-8.7	16.8	24.3	-1.7
CACZ09	13.5	29.5	84.2	19.0	-14.8	0.1	76.3	-9.1	-6.2	17.3	33.8	0.3
CACZ10	14.4	25.7	75.9	18.8	-11.8	-0.4	65.6	-6.9	-3.6	14.5	28.5	1.4
CACZ11	8.6	8.6	53.5	14.3	-20.2	-0.3	52.5	0.7	-9.2	9.6	23.1	5.9
CACZ12	11.8	19.8	57.5	22.2	-23.0	-2.8	57.0	-1.2	-10.2	9.4	23.1	5.5
CACZ13	13.0	19.9	58.4	20.9	-14.2	-3.1	57.4	-1.6	-3.8	10.5	23.0	6.9
CACZ14	9.9	20.2	60.6	16.8	-11.3	3.0	50.3	-2.5	-3.2	12.9	32.0	3.3
CACZ15	15.3	28.2	86.1	18.0	-2.5	2.8	77.0	-1.0	5.3	17.7	41.0	7.6
CACZ16	9.5	24.1	48.5	26.1	-41.5	-2.1	41.6	-4.6	-18.6	11.8	27.7	3.6

Table 9. School Classroom Wing Savings Comparisons

	Demand Controlled Ventilation + EC				Heat Pump Heat Recovery				Enthalpy Exchanger			
	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %
CACZ01	-3.7	3.9	99.3	7.4	-88.1	-13.9	81.3	-30.2	-88.2	-13.0	74.6	-30.0
CACZ02	4.0	6.5	95.7	9.0	-32.5	-4.0	76.1	-11.8	-29.4	3.4	78.7	-6.2
CACZ03	-0.3	8.2	99.0	6.5	-57.2	-7.4	84.0	-23.7	-57.1	2.9	77.0	-17.3
CACZ04	6.0	11.1	99.3	10.4	-33.2	-3.8	84.6	-14.2	-28.8	10.7	85.3	-3.9
CACZ05	-2.3	1.6	100.0	1.0	-49.5	-2.6	89.2	-19.5	-50.2	2.9	92.3	-16.5
CACZ06	-0.2	14.6	100.0	2.1	-45.6	-0.9	100.0	-38.6	-44.0	12.0	75.0	-35.2
CACZ07	3.4	11.9	100.0	5.8	-41.8	-9.1	100.0	-32.5	-39.0	3.7	100.0	-26.8
CACZ08	5.7	15.2	100.0	7.3	-27.6	-2.9	95.8	-23.6	-23.0	13.6	95.8	-17.2
CACZ09	6.5	17.6	100.0	8.3	-26.8	1.0	100.0	-22.4	-21.4	16.6	100.0	-15.4
CACZ10	8.6	14.0	100.0	9.8	-19.4	0.9	87.5	-16.0	-14.5	13.1	92.2	-10.1
CACZ11	6.7	7.7	93.7	10.8	-21.9	-1.2	76.4	-6.9	-17.1	4.9	74.5	-1.7
CACZ12	6.8	9.3	97.3	11.1	-26.4	-3.6	75.9	-10.5	-21.5	5.1	76.5	-3.7
CACZ13	9.5	10.1	98.1	11.7	-16.8	-3.0	77.0	-7.5	-11.0	7.4	78.2	0.6
CACZ14	7.2	12.6	93.3	10.7	-13.8	3.2	66.7	-8.9	-9.6	10.3	80.5	-3.9
CACZ15	13.1	17.4	100.0	13.7	-6.2	2.2	100.0	-5.1	1.2	14.1	100.0	2.8
CACZ16	3.9	11.9	84.9	18.7	-36.6	-2.4	65.3	-14.3	-27.4	7.6	71.3	-5.4

Table 10. School Auditorium Savings Comparisons

	Demand Controlled Ventilation + EC				**Heat Pump Heat Recovery**				**Enthalpy Exchanger**			
	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %	Elec. Energy, %	Elec. Dmd, %	Gas, %	Total, %
CACZ01	-1.7	-1.3	88.7	45.3	-233.9	-126.5	90.9	-28.4	-111.1	-54.9	11.2	-28.3
CACZ02	20.9	41.7	86.2	45.8	-53.2	-6.6	81.2	0.3	-22.2	17.8	20.1	10.9
CACZ03	3.7	32.7	91.5	40.6	-117.7	-22.0	92.4	-12.3	-68.6	4.6	11.2	-4.6
CACZ04	24.2	51.7	90.1	51.0	-48.9	-7.0	88.3	-3.8	-20.6	22.4	14.0	13.3
CACZ05	7.0	35.1	93.6	36.6	-76.6	-11.2	93.7	-11.1	-52.9	9.8	6.8	-1.4
CACZ06	13.7	46.4	98.7	30.1	-51.0	-1.9	98.3	-24.7	-37.5	16.8	3.6	-17.8
CACZ07	21.4	46.9	99.7	37.1	-40.6	-4.5	99.7	-16.7	-27.2	19.8	3.7	-4.3
CACZ08	28.4	52.3	96.4	38.6	-24.2	-2.0	96.2	-12.1	-9.6	25.0	6.0	1.6
CACZ09	26.6	52.2	97.7	36.7	-17.0	2.1	96.4	-7.2	-4.6	27.0	10.4	5.1
CACZ10	30.1	49.5	96.2	38.2	-13.0	4.7	95.4	-3.7	0.3	26.5	9.2	7.9
CACZ11	21.7	37.1	82.8	39.9	-30.3	0.1	75.9	3.6	-8.8	20.9	21.8	14.1
CACZ12	23.6	43.3	85.5	45.0	-34.9	-4.5	80.7	1.0	-11.9	18.6	20.0	12.0
CACZ13	26.7	43.9	86.3	43.4	-19.8	-1.6	81.7	1.6	-1.3	21.3	19.4	15.1
CACZ14	26.1	45.7	87.2	37.9	-17.7	9.9	76.1	-0.1	0.7	27.9	26.3	10.2
CACZ15	33.4	52.9	98.2	38.0	0.9	8.4	97.6	3.2	13.5	32.0	11.0	17.4
CACZ16	21.8	43.2	76.7	48.3	-75.3	1.0	62.3	-4.1	-27.5	19.5	30.6	5.7

Payback Periods

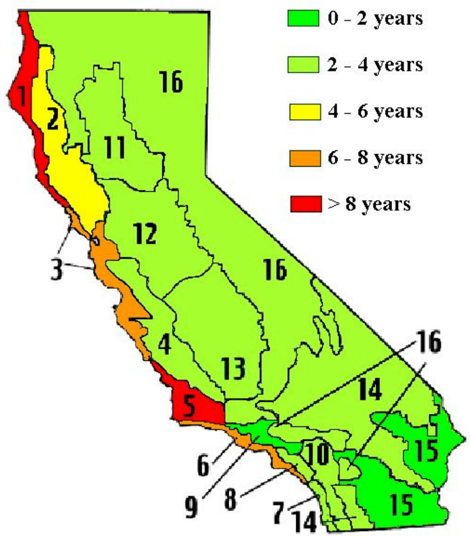
Yearly payback periods associated with the ventilation strategies are highly dependent upon first costs. Section II describes the assumptions used to estimate first costs for the different technologies. All payback periods assume a retrofit application. DCV requires the lowest first costs because of lower installation time and equipment costs, followed by the enthalpy exchanger and then the heat pump heat recovery unit.

Table 11 shows payback periods for all building types and locations throughout California for DCV+EC.

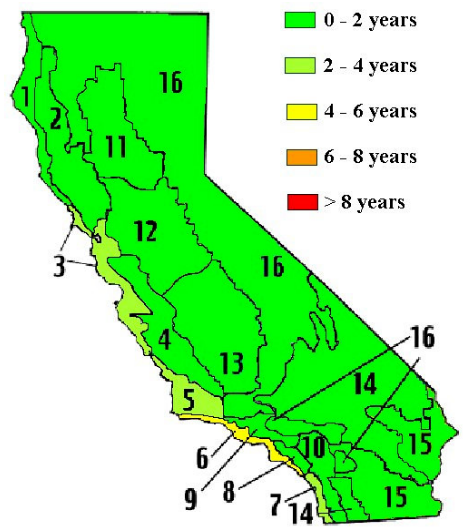
Figure 14 shows the payback periods on a California map for four of the buildings covering the range of results. The payback periods associated with DCV are very attractive for most all applications throughout California. As expected, the lowest payback periods occur in the more extreme climates and for buildings with a lower ratio of average to peak occupancy. The payback periods are significantly higher in the coastal climates because of significantly lower cooling and heating requirements and greater economizer opportunities. Therefore, less opportunity for savings with DCV control exists. The payback periods are also significantly higher for the office, restaurant, library, and classroom because of higher average occupancy.

Table 11. Payback Periods for DCV + EC (years)

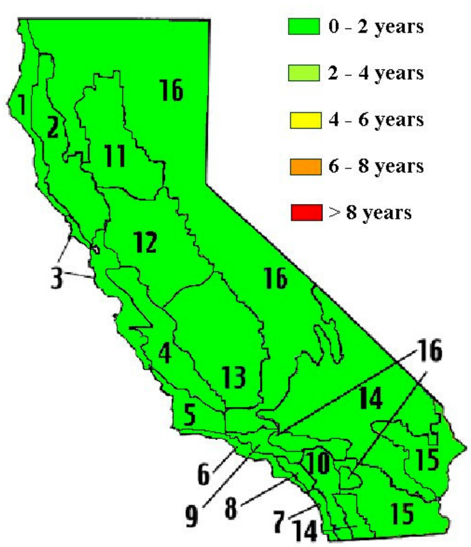
	Office	Restaurant	Retail Store	Library	Gym	Classroom	Auditorium
CACZ01	8.0	1.4	0.6	6.8	1.0	5.2	0.4
CACZ02	5.0	0.5	0.6	9.6	1.2	2.3	0.5
CACZ03	6.8	2.1	1.0	7.6	1.6	4.0	0.6
CACZ04	3.0	1.1	0.6	7.4	0.8	1.8	0.4
CACZ05	17.9	2.9	1.8	39.5	2.2	24.2	0.7
CACZ06	6.0	4.0	1.7	13.9	2.0	9.0	0.9
CACZ07	3.9	3.4	1.5	13.1	1.9	3.9	0.8
CACZ08	3.7	0.9	0.9	11.7	1.2	2.1	0.7
CACZ09	1.6	1.4	0.8	9.8	1.0	1.6	0.6
CACZ10	3.4	1.1	0.6	8.3	1.0	1.4	0.6
CACZ11	3.1	1.0	0.7	9.2	1.3	1.6	0.5
CACZ12	3.2	1.0	0.6	7.0	0.8	1.6	0.4
CACZ13	2.9	0.8	0.5	6.3	0.8	1.3	0.4
CACZ14	2.5	0.8	0.6	8.2	1.0	1.2	0.5
CACZ15	1.9	0.6	0.3	4.4	0.9	0.9	0.4
CACZ16	3.5	0.6	0.4	2.8	0.9	1.0	0.4



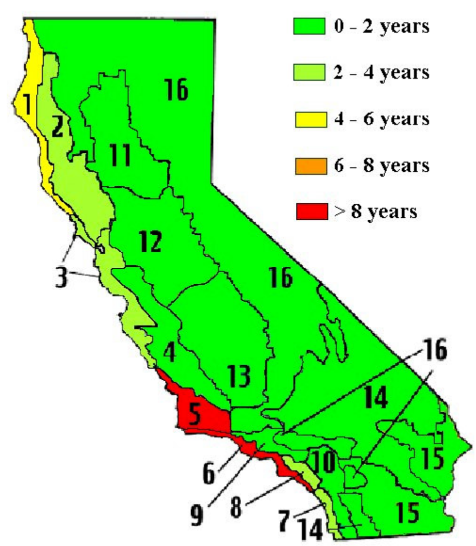
Office



Restaurant



Retail Store



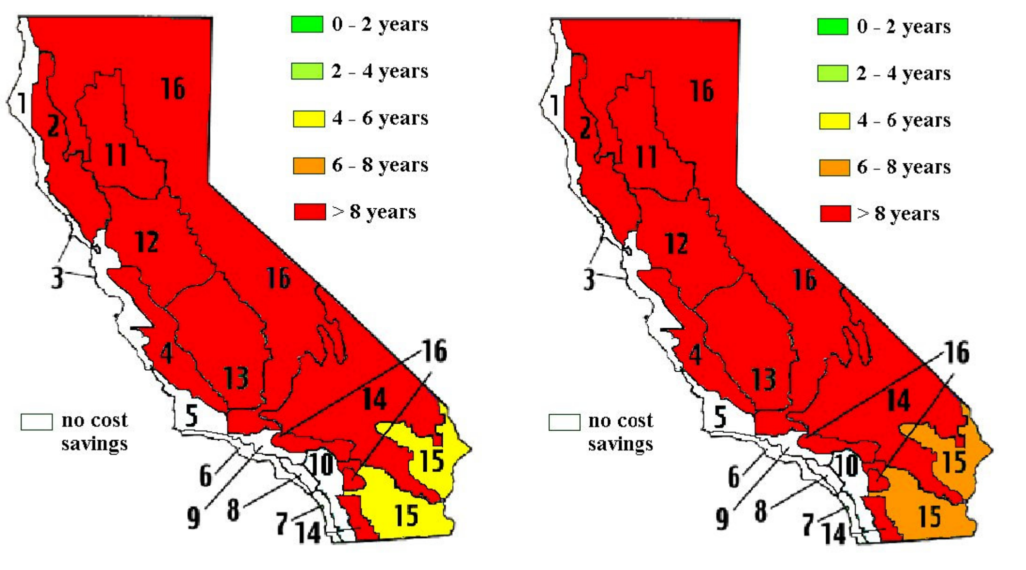
School Classroom

Figure 14. Sample Payback Periods for DCV + EC

Table 12 shows payback periods for enthalpy exchangers as a retrofit in all the building types and locations throughout California. Figure 15 shows results for four of the buildings superimposed on a map. Paybacks for the enthalpy exchanger are typically greater than 7 years for most areas of California, except for some building types in climate zone 15. The payback periods were determined assuming that the primary equipment was not resized with the addition of the enthalpy exchanger (i.e., it's a retrofit). The paybacks would be lower for new installations where the primary cooling and heating equipment were downsized in response to lower ventilation loads.

Table 12. Payback Periods for Enthalpy Exchangers (years)

	Office	Restaurant	Retail Store	Library	Gym	Classroom	Auditorium
CACZ01	-	-	-	-	-	-	-
CACZ02	-	19.0	36.5	-	31.7	-	21.8
CACZ03	-	-	-	-	-	-	-
CACZ04	-	17.6	28.9	-	12.4	-	17.0
CACZ05	-	-	-	-	-	-	-
CACZ06	-	-	-	-	-	-	-
CACZ07	-	-	-	-	-	-	-
CACZ08	-	-	-	-	-	-	-
CACZ09	-	-	-	-	-	-	-
CACZ10	-	-	-	-	-	-	27.9
CACZ11	-	10.1	12.6	-	15.2	-	13.4
CACZ12	-	14.5	20.5	-	17.0	-	16.9
CACZ13	-	8.9	10.0	17.6	11.8	-	11.2
CACZ14	-	13.9	25.1	-	25.1	-	18.4
CACZ15	23.8	4.9	5.0	13.6	7.3	12.0	7.0
CACZ16	-	11.1	38.3	-	32.3	-	46.7



Restaurant and Retail Store **Auditorium and Gymnasium**

Figure 15. Sample Payback Periods for Enthalpy Exchangers

The heat pump heat recovery system does not provide cost savings for many locations in California. Furthermore, for the locations where savings do occur the payback periods are not reasonable. Figure 16 shows the best case results for this technology. In addition to smaller savings, first costs for the heat pump are significantly higher than for the other two ventilation strategies. Savings only occur with very extreme ambient conditions.

The paybacks would be somewhat smaller for new installations than for retrofits because the primary air conditioning and heating equipment could be downsized. However, it is not expected that it could be competitive with an enthalpy exchanger or DCV for new installations in California.

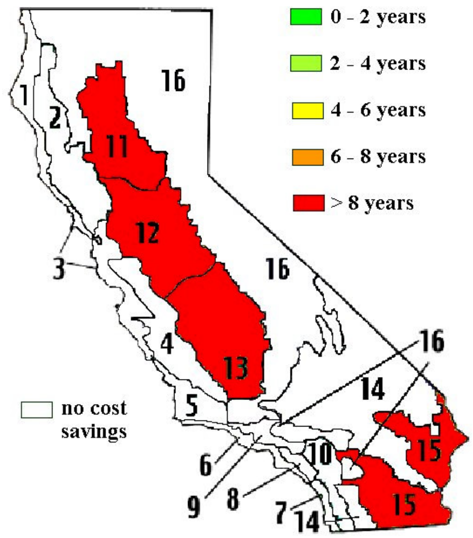


Figure 16. Sample Payback Periods for Heat Pump Heat Recovery

Impact of Occupancy

The savings associated with each ventilation strategy are strongly dependent upon the peak occupant density and average occupancy schedules. The peak occupant density is important because it establishes the fixed ventilation requirement for the base case, HPHR, and HXHR systems. The average occupancy is important for DCV because ventilation varies indirectly with occupancy. For the default simulations, the occupancy schedules and peak occupant densities were assumed based on the LBNL study (Huang, et al. 1990 & Huang, et al. 1995). Average hourly occupancy values were assumed in relation to a daily average maximum occupant density (people per 1000 ft^2).

Figure 17, Figure 18 and Figure 19 show savings potential for three different peak occupant densities (7, 30, and 150 people per 1000 ft^2) as a function of average occupancy relative to the peak for the three ventilation strategies for the office building prototype in CZ 15. Percent savings decrease as the average-to-peak occupancy ratio increases for all three ventilation strategies. The average occupancy was assumed to be constant for all occupied hours of the day and days of the year. For DCV, as the relative occupancy approaches the peak value, the opportunity for modulating the outside air damper in response to zone CO_2 diminishes. At 100% peak occupancy, DCV does not modulate the damper below the fixed ventilation requirement and the savings are zero. The savings for DCV also increase with peak occupant density. This is because the ventilation load associated with the base case having fixed ventilation increases with occupant density due to an increase in the required ventilation rate. Thus, there is a greater opportunity for reducing the ventilation load.

The heat pump and enthalpy exchanger systems exhibit similar trends. The energy recovery opportunites are greater for the higher ventilation rates associated with the higher peak occupancies. For a given peak occupancy, the sensitivity of savings to average occupancy is less than for the DCV case. The primary impact of average occupancy on operating costs for the base case, heat pump, and enthalpy exhanger systems is due to increased internal gains. At lower internal gains associated with lower average occupancy, the ventilation cooling load is a larger fraction of the total cooling load and the relative savings for energy recovery increase.

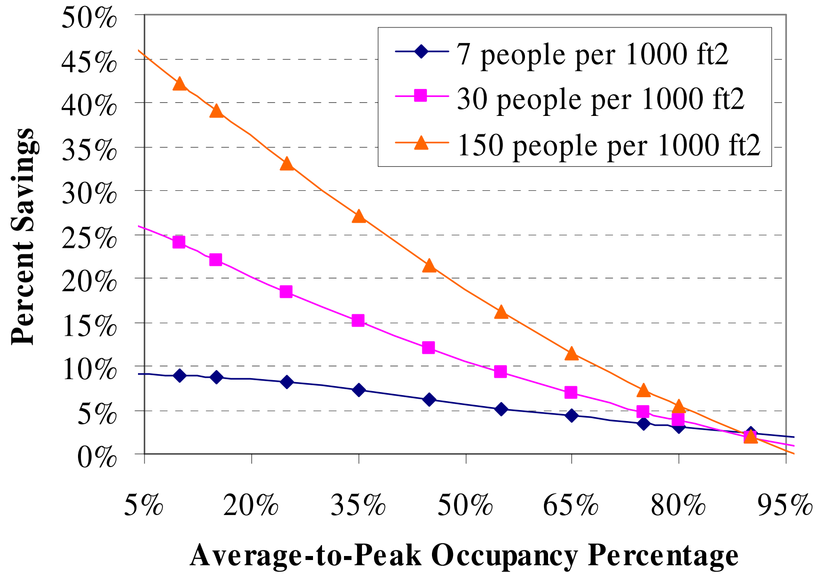


Figure 17. DCV+EC Percent Savings vs. Average-to-Peak Occupancy Percentage for Office in CZ 15

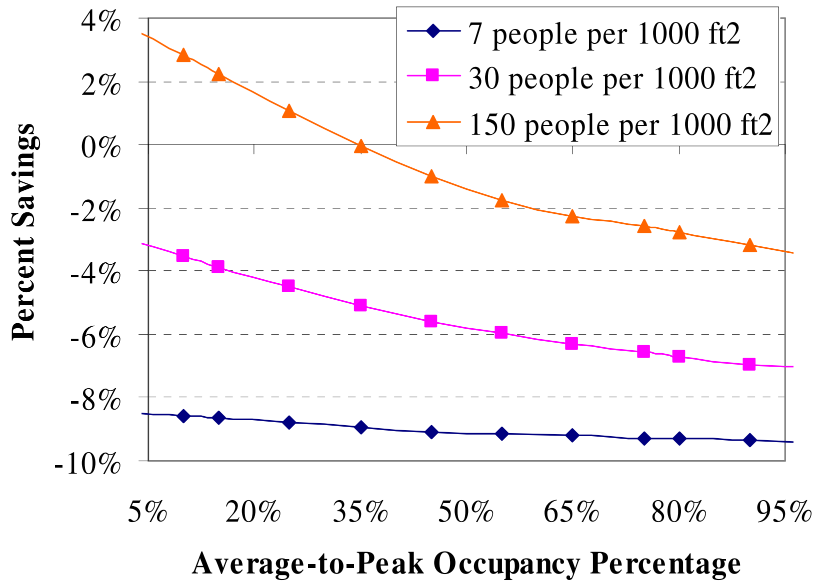


Figure 18. HPHR Percent Savings vs. Average-to-Peak Occupancy Percentage for Office in CZ 15

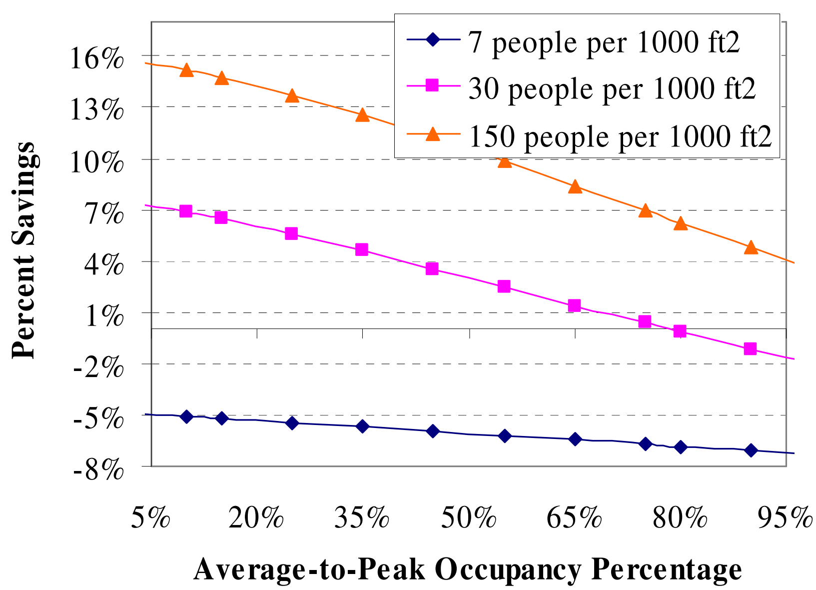


Figure 19. HXHR Percent Savings vs. Average-to-Peak Occupancy Percentage for Office in CZ 15

Impact of Exhaust Fan Efficiency

The heat pump (HPHR) and enthalpy exchanger (HXHR) ventilation strategies both require a fan in the exhaust air stream to overcome additional pressure losses. In some applications, an additional ventilation fan may also be necessary. The default HPHR fan power was based upon measurements from a commercial unit having only an exhaust fan and is consistant with a fan/motor efficiency of 15% and a static pressure loss for the wheel media or heat exchanger of 0.64 inches of water.

Figure 20 shows the effect of the exhaust/ventilation fan power on the relative savings for the HPHR and HXHR systems for July 19 in CZ 16. A value of 0.2 watts per cfm is representative of a system having only an exhaust fan, but with improved fan/motor efficiency. A value of 1.0 watts per cfm is representative of a system having both an exhaust and ventilation fan with the default fan/motor efficiency. The fan power can make the difference between positive and negative savings for the HXHR and HPHR systems. Although lowering the fan power for the HPHR system does not result in positive savings for this case, it does increase the number of situations (building types / climate zones) where the HPHR system yields positive savings. The lower fan power for the HXHR does not lead to payback periods that are competitive with DCV+EC for the systems considered.

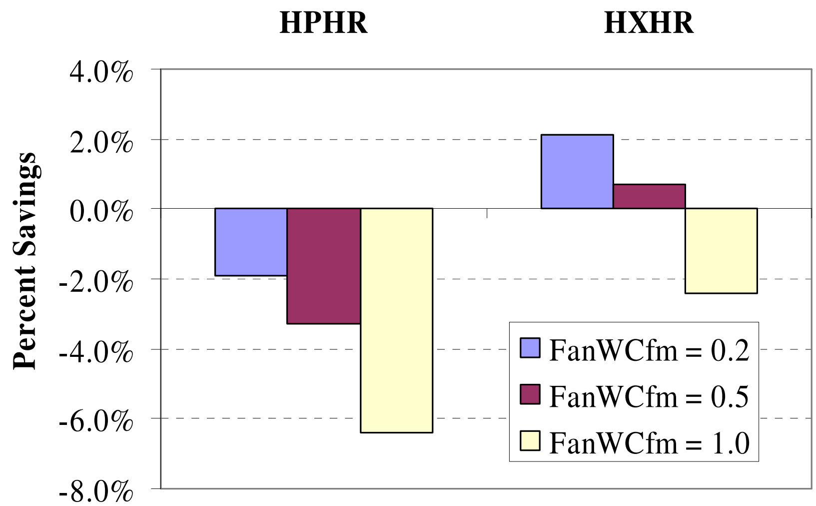


Figure 20. Annual Percent Savings for Different Exhaust/Ventilation Fan Powers – Office, CZ15

Impact of Economizer for HPHR and HXHR Systems

One of the reasons that DCV systems have greater cost savings than HPHR and HXHR systems in California is that these alternative technologies do not incorporate economizers. Although an "economzer mode" for the HPHR and HXHR systems involves turning the units off when the ambient temperature is below the return air temperature, the ventilation flowrate is fixed at the value necessary to satisfy ASHRAE 62-1999. For the DCV systems, the economizer allows the use of 100% outside air and significantly greater "free cooling" can be achieved.

In order to evaluate the penalty associated with the loss of free cooling, a differential enthlapy economizer was implemented in combination with the HPHR and HXHR systems. When the economizer is enabled, the ventilation heat pump or enthalpy exchanger is off and the outside air damper is controlled to meet a mixed air temperature set point of 55 F or is fully open. Two different implementations for the economizer were considered: 1) the ventilation and exhaust air are assumed to flow through the heat pump or enthalpy exchanger in economizer mode, so that the exhaust fan must operate and 2) the ventilation and exhaust flows are assumed to bypass the heat pump or enthalpy exchanger in economizer, so the exhaust fan is turned off. The first implementation would only require a controllable return damper, whereas the second implementation would require controllable ventilation, exhaust, and return dampers but would require less fan power.

Figure 21 and Figure 22 show example comparisons of the HPHR and HXHR systems with and without a flow-through economizer for a mild California climate (CZ 06) and a hot climate (CZ 15) for the restaurant. For these examples, the exhaust fans operate and the return air damper is closed when the economizer is enabled. These figures also include results for DCV both with and without an economizer.

In the mild climate, the savings are negative for both the HXHR and HPHR technologies indicating that the base case with a differential economzier has lower utility costs. This is due to the extra power associated with running the exhaust fan for the HXHR system and the compressor and exhaust fan for the HPHR system. The use of an economizer does significantly reduce the costs, but savings are still negative. Savings for DCV without the use of an economizer are also negative. For the hot climate, savings associated with both the HPHR and HXHR technologies are positive. The use of an economizer increases the savings, but has a smaller effect than for the mild climate.

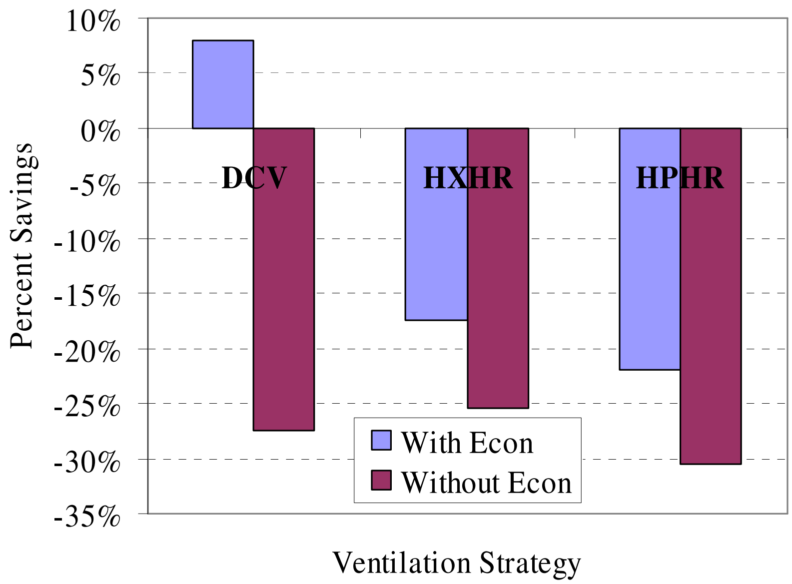


Figure 21. Flow-Through Economizer Savings for the Restaurant in CZ 06

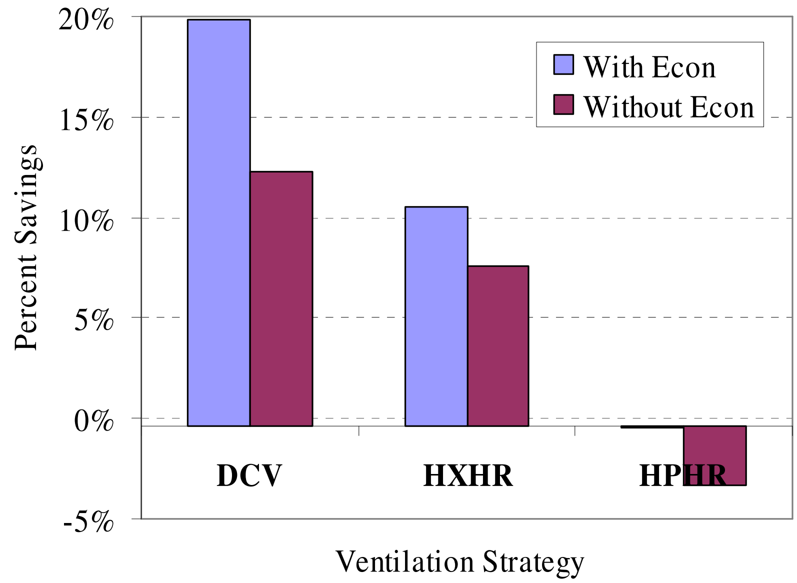


Figure 22. Flow-Through Economizer Savings for the Restaurant in CZ 15

Figure 23 and Figure 24 show example results for the bypass economizer. In this case, the ventilation bypasses the heat pump or enthalpy exchanger and the exhaust fan is off during economizer operation. The performance of the HXHR and HPHR systems improve in both climates for the bypass economizer, but is still not competitive with DCV.

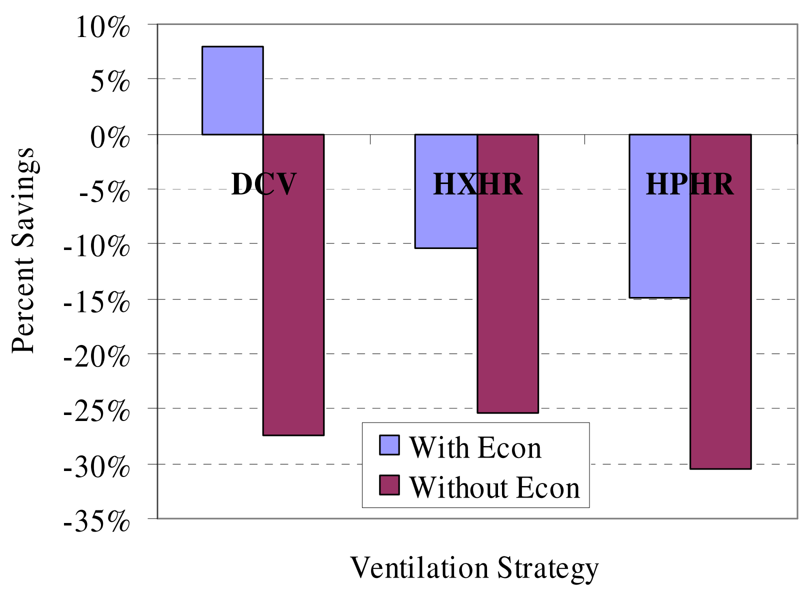


Figure 23. Bypass Economizer Savings for the Restaurant in CZ 06

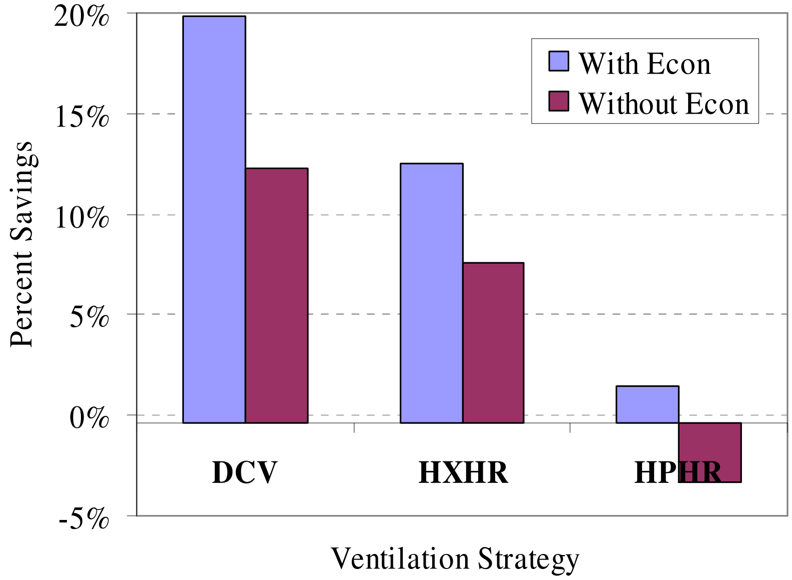


Figure 24. Bypass Economizer Savings for the Restaurant in CZ 15

Zone Humidity Comparisons

One of the advantages of any of the three alternative technologies is lower humidity levels in the zones during the cooling season for humid climates. DCV reduces moisture gains due to ventilation as a result of reduced ventilation air flow. The heat pump heat recovery unit and enthalpy exchanger remove moisture from the ventilation stream as part of the overall energy recovery.

Figure 25, Figure 26 and Figure 27 compare occupied period zone relative humidities for the month of July in Houston for the restaurant, office and auditorium. The results are presented as histograms of relative humidity between 30% and 80%. Relative humidities greater than about 60% are outside of the ASHRAE recommended range of comfort. These zone relative humidities were calculated by controlling the zone temperature to 75 F.

For the office, Figure 25 shows that zone conditions remained within the comfort range for the base case and three alternative ventilation strategies. All three alternative ventilation technologies resulted in reduced humidity levels when compared with the base case. DCV resulted in the lowest zone humidity levels, followed by the enthalpy exchanger and then the ventilation heat pump system.

Figure 26 and Figure 27 show similar trends for the restaurant and auditorium. However, the zone humidity levels were much higher for the strategies having fixed ventilation rates (base case, HPHR, and HXHR) because of the high design occupant densities. Both the base case and the HPHR systems had a significant number of hours with relative humidities greater than 60%. In actual operation, the zone set point would be lowered below 75 F in order to achieve zone humidities within the comfort area. The DCV system had significantly lower humidity ratios than the other technologies for the auditorium because this application has low average occupancies.

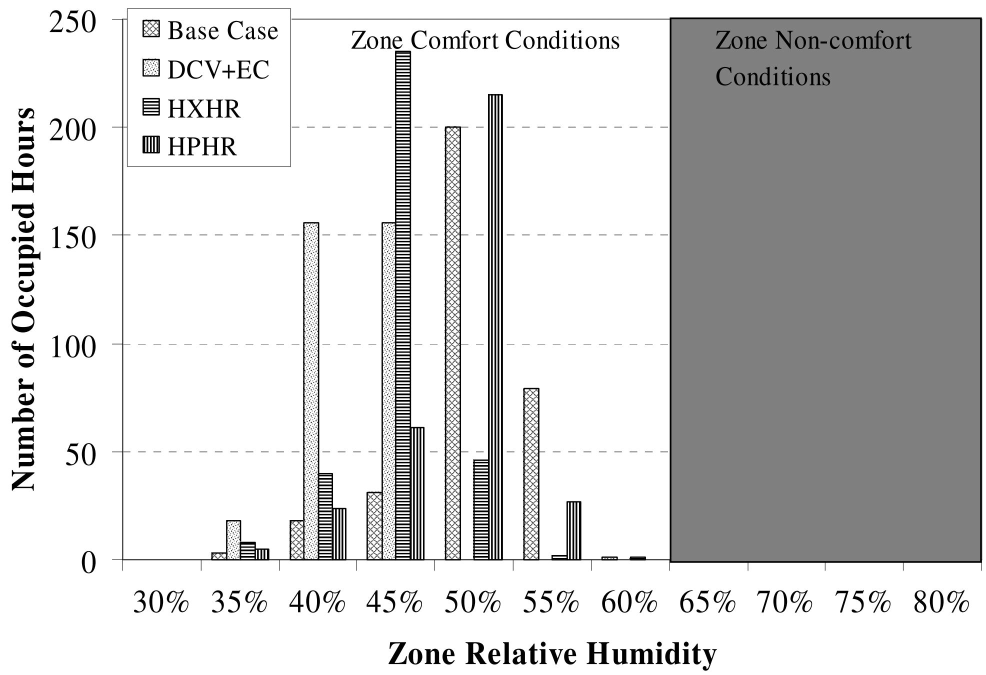


Figure 25. Occupied Hours of Zone Relative Humidity for the Office in Houston

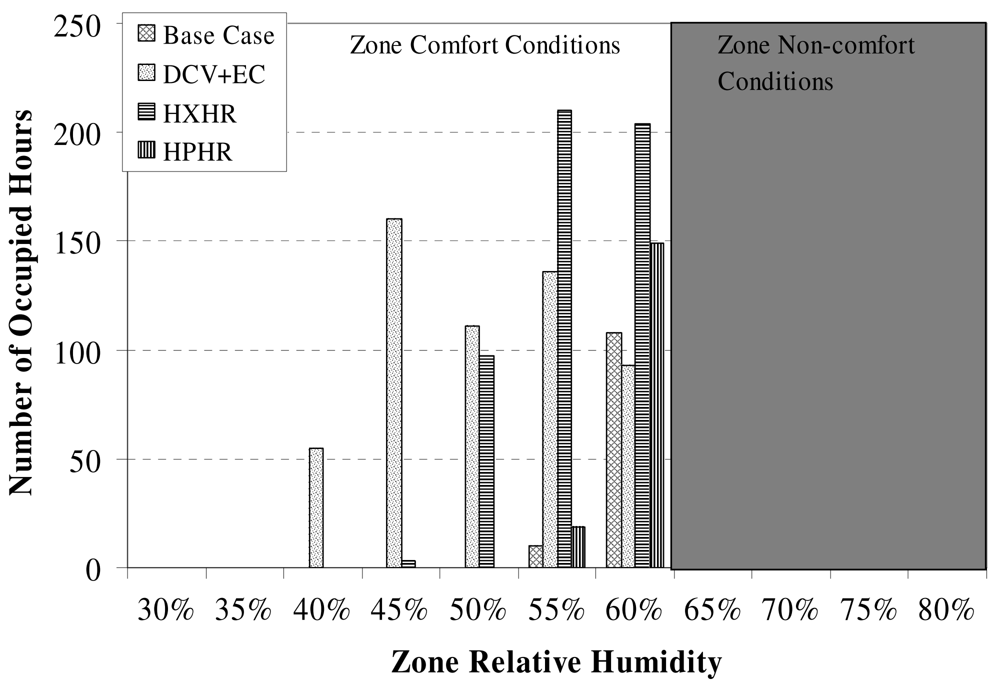


Figure 26. Occupied Hours of Zone Relative Humidity for the Restaurant in Houston

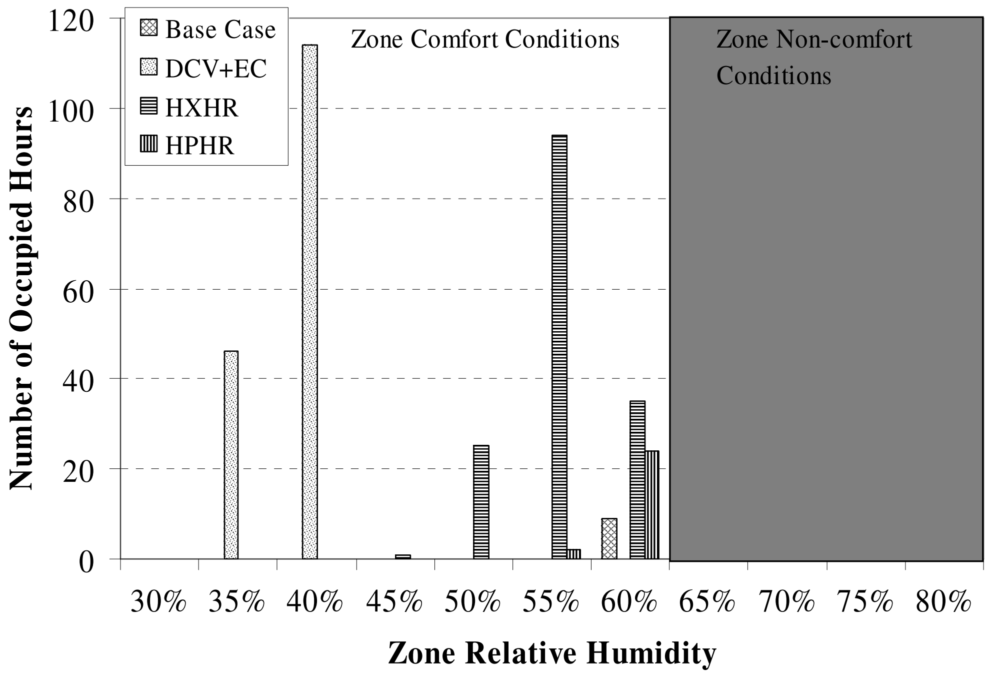


Figure 27. Occupied Hours of Zone Relative Humidity for the Auditorium in Houston

New Building Applications

Additional cost savings are possible for new applications with systems employing enthalpy exchangers or heat pump heat recovery. The use of energy recovery leads to reduced primary equipment loads and an opportunity to downsize the primary RTUs. Operating cost savings may also increase for these systems in new applications compared to retrofit applications due to a decrease in primary RTU on/off cycling resulting from the downsizing.

In order to assess the impact of RTU resizing on savings, four building types in two climate zones were investigated: the office, restaurant, retail store and school auditorium in CACZ 06 and 15. These combinations cover mild and hot climate zones with a large variability in peak occupant density and occupancy schedules relative to the peak occupant density.

A rate of return for each case was calculated for comparisons with the base case and DCV+EC. RTU equipment cost savings were calculated using an installed cost of $1000 per ton of cooling. Reductions in primary heater equipment costs were not considered. For DCV+EC, the RTU can not be downsized for new designs.

Figure 28 through Figure 35 show cumulative rates of return for the different cases as a function of year after the retrofit. The simple payback period occurs at the point where the rate of return becomes positive. Several conclusions can be made from these results, including: 1) rates of return are higher in the hotter climate and for the buildings having higher peak occupancy (e.g, the auditorium versus the office), 2) the HXHR and HPHR systems do not have positive rates of return in the moderate climate, 3) in the hotter climate, the enthalpy exchanger results an immediate rate of return (immediate payback) due to RTU equipment cost savings, 4) although the rates return for the DCV+EC start out negative (due to the initial investment), they surpass the enthalpy exchanger rates of return within a short time period, and 5) the HPHR system is not competitive with the other technologies.

Overall, the conclusions do not change for new building applications. Demand-controlled ventilation has better overall economics than the other energy recovery technologies for the systems and conditions considered in this study. More detailed results are given in Appendix C.

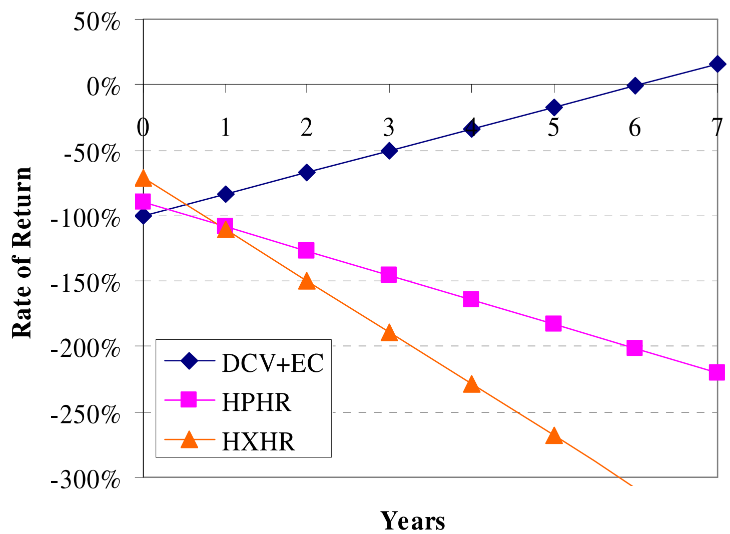


Figure 28. Cumulative Rate of Return for Office in CACZ 06

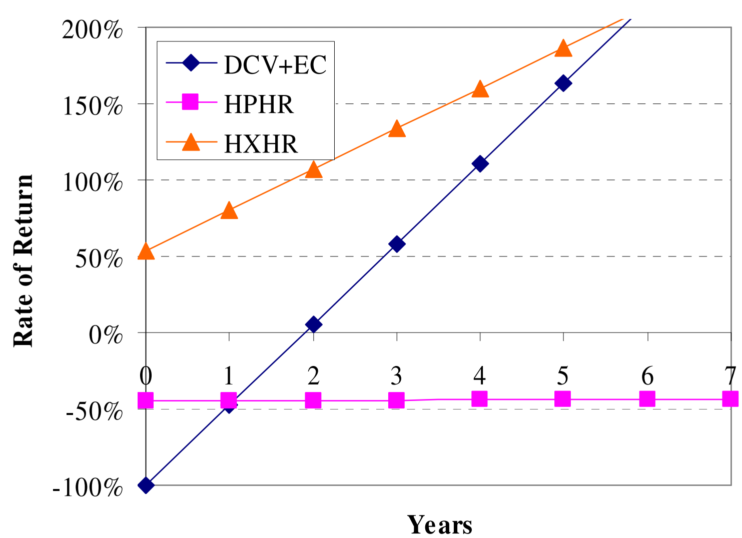


Figure 29. Cumulative Rate of Return for Office in CACZ 15

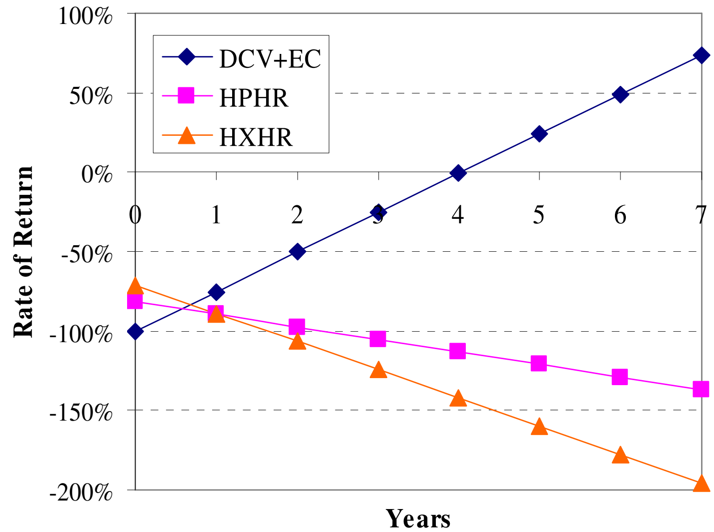


Figure 30. Cumulative Rate of Return for Restaurant in CACZ 06

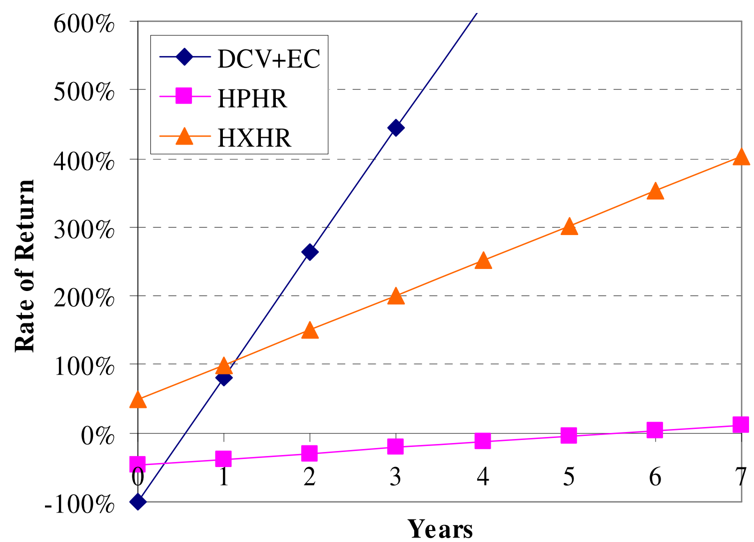


Figure 31. Cumulative Rate of Return for Restaurant in CACZ 15

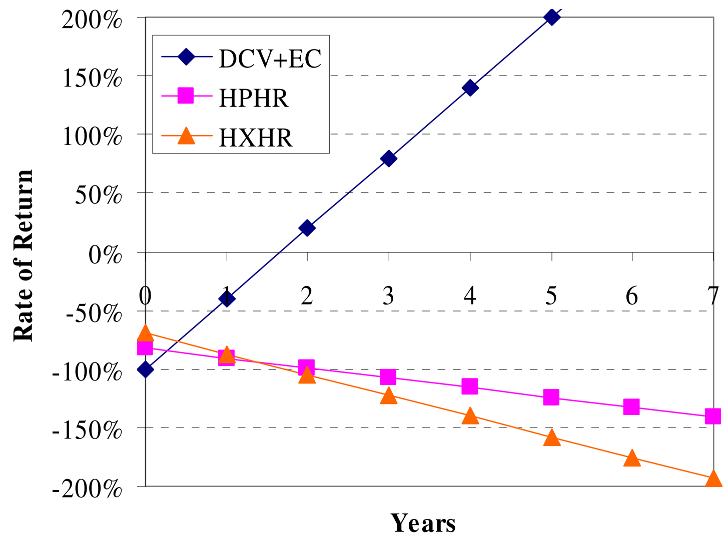


Figure 32. Cumulative Rate of Return for Retail Store in CACZ 06

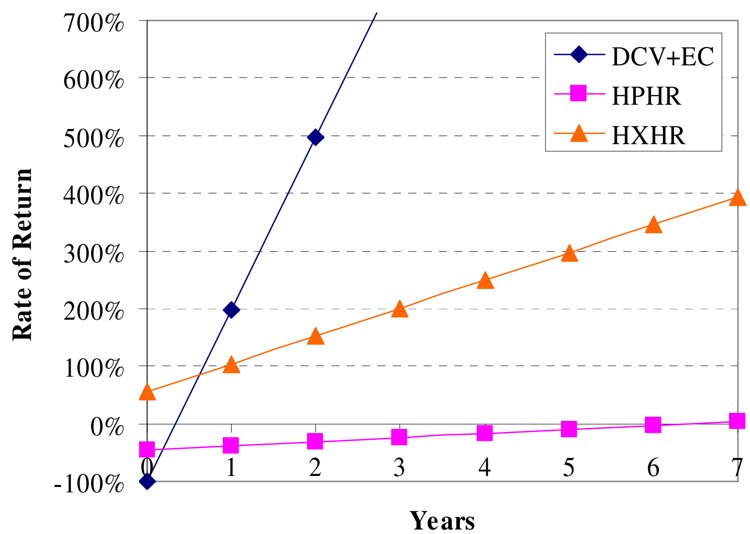


Figure 33. Cumulative Rate of Return for Retail Store in CACZ 15

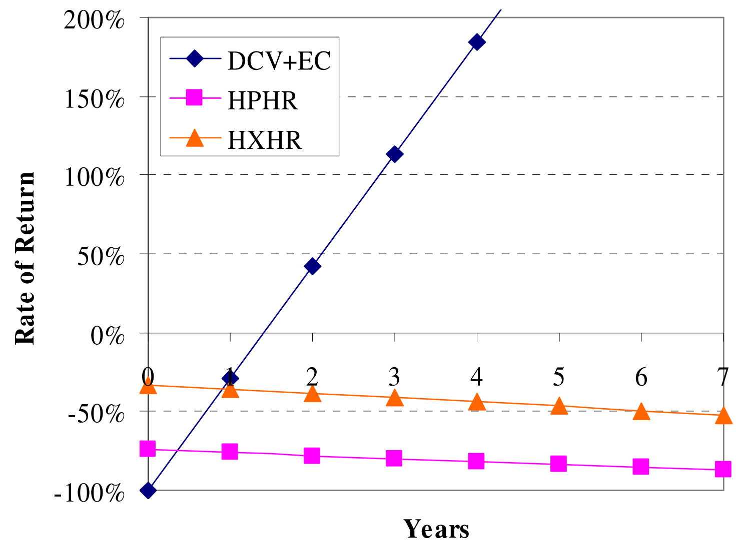


Figure 34. Cumulative Rate of Return for Auditorium in CACZ 06

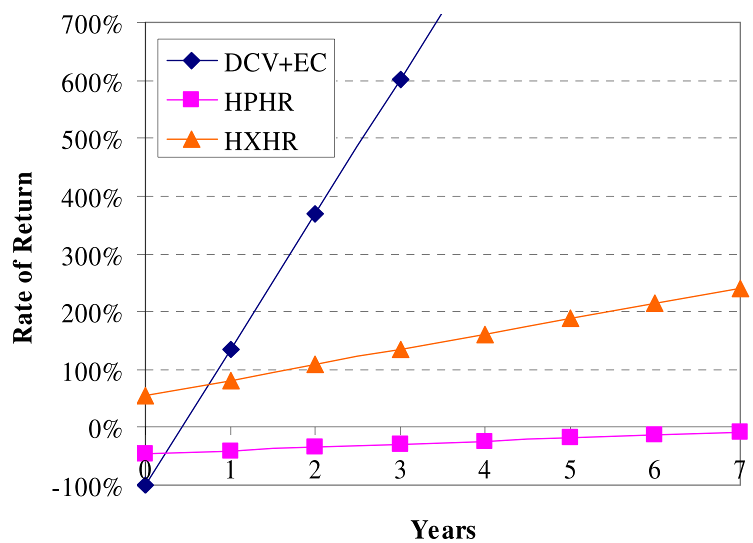


Figure 35. Cumulative Rate of Return for Auditorium in CACZ 15

IV. DCV FIELD TESTING

DCV Field Sites

For evaluation of DCV, field sites were established for three different building types in two different climate zones within California. The building types are: 1) McDonalds PlayPlace® areas, 2) modular school rooms, and 3) Walgreens drug stores. In each case, nearly duplicate test buildings were identified in both coastal and inland climate areas.

This section provides a brief overview of these field sites. A detailed description of the field test sites and the data collection system is included in Deliverable 3.1.1a (2003).

The PlayPlace areas are isolated from the main dining area and have separate packaged rooftop HVAC unit(s). Heating is provided by natural gas burners. Two restaurants sites are located approximately 15 miles apart in the San Francisco Bay area (south of Oakland and north of San Jose). Two other restaurant sites are in the Sacramento area.

The modular schoolrooms are typical of thousands employed throughout California and the United States. They use a single sidewall mounted packaged heat pump system. Two schoolrooms are located in Oakland and two are in Woodland, just east of Sacramento.

The drug stores selected for this study are larger than the other field sites and use five rooftop units that service the store and pharmacy areas. Due to the larger number of HVAC units at the Walgreens sites, only one store in each climate type is being monitored. One store is near Riverside and the other is in Anaheim.

The two alternative control strategies compared were DCV with economizer control (DCV On) and economizer cooling only (DCV Off). With the DCV On strategy, the return air CO_2 set point was 800 ppm_v. When the return air CO_2 concentration was below the set point, the outdoor air ventilation damper was fully closed. Otherwise, the Honeywell controller provided feedback control of the damper position. For the DCV Off mode, a minimum damper position was set so as to provide the required outdoor airflow as specified in ASHRAE Standard 62-1999 (ASHRAE, 1999). The fixed damper position that satisfies the standard was estimated to be 40% for the McDonalds and the modular schools and 20% open for the Walgreens stores. However, field airflow measurements at one McDonalds store indicates that the actual total supply airflow varies significantly with damper position. This impacts the actual amount of ventilation air provided.

The field measurements for HVAC equipment included electric power, integrated electrical energy, digital control signals for the gas valve and supply fan, ambient, return, and mixed air temperature and humidity, supply air temperature, and return air CO_2 concentration.

The power is calculated from voltage and current readings for each unit (fans plus compressor). For the Bradshaw Road and Milpitas sites that have two rooftop units, only direct power measurements from one of the units were available, but they are duplicate systems. Operation of the second rooftop unit was monitored via the digital control signals indicating fan, cooling or heating being on. Since the modular school sites use a single phase electrical power connection, separate monitoring of the total unit and compressor power is performed.

Data were collected every five minutes and downloaded to a server on a daily basis using a cell phone. A website provided direct access to the data. A screening analysis program was used to check for erroneous data and compute hourly averages.

Installation at the field test sites began in late 2000 with installation, checkout and debugging finished by the end of 2001 for the McDonalds and modular schoolroom sites. The Walgreens store installation and debugging continued into 2002.

Comparison Methodologies

The costs for heating associated with the field sites are relatively small compared to the cooling costs and only cooling season results are presented in this report. Heating season results are described by Lawrence and Braun (2003). The cooling season results are presented in this report using the different approaches described below.

Direct side-by-side comparisons – Nearly identical sites were chosen in the northern California climates to allow direct side-by-side comparisons for the same time periods. As a check on the differences between sites, it is important to also compare energy use with both sides operating in the same mode (e.g., DCV On or DCV Off).

Correlated daily energy usage – This approach involves comparison of average daily energy use for heating or cooling at the same site. Total daily energy usage was correlated as a function of average ambient temperature for different time periods when the DCV was on and off. Separate correlations were developed for DCV On and Off and then used to compare energy use for a given daily ambient condition or over a period of time (e.g., cooling or heating season).

Calibrated simulation – Field site information and data were used to develop VSAT simulations for the field sites. The field measurements were then compared with VSAT predictions using short-term data for validation purposes and annual simulations were performed to evaluate savings and economic payback. Lawrence and Braun (2003) present additional comparisons of energy usage based upon hourly models that were derived from the data.

In addition, CO_2 levels in the zone were compared for DCV and fixed ventilation.

Field Results for McDonalds PlayPlace Areas

Side-by-Side Energy Use Comparisons

Variations in the DCV control settings were made at the Milpitas and Castro Valley sites in the San Francisco area to allow side-by-side comparisons. Figure 36 shows daily energy usage for cooling (compressor + fan energy) for a time period where DCV was off for both sites. The Castro Valley site had slightly higher energy consumption (82.8 kW-hr per day) compared to the Milpitas site (80.0 kW-hr per day), a difference of about 3.5%.

Figure 37 shows side-by-side comparisons of daily cooling energy usage for DCV On and DCV Off at the two sites during a three-week period. The strategies were alternated between the two sites, but the savings for DCV On were nearly the same regardless of which sites were on and off. Average measured daily energy savings for DCV On was about 14% for this time period.

Bay Area McDonalds - Both Stores with DCV Off

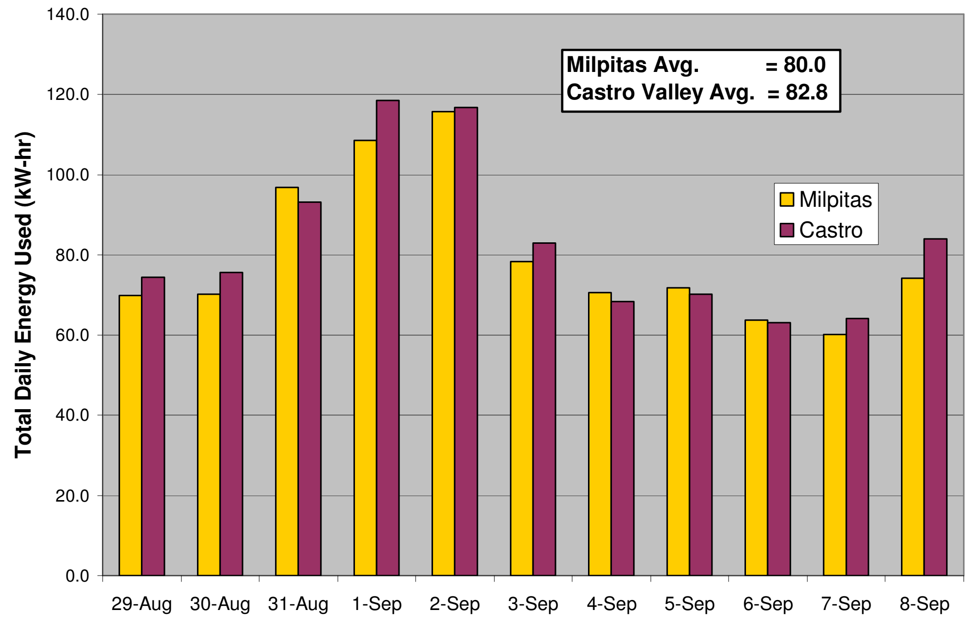


Figure 36. Cooling Energy Use for DCV Off at Bay Area McDonalds

Bay Area McDonalds Side-by-Side Comparison (August 2002 Data)

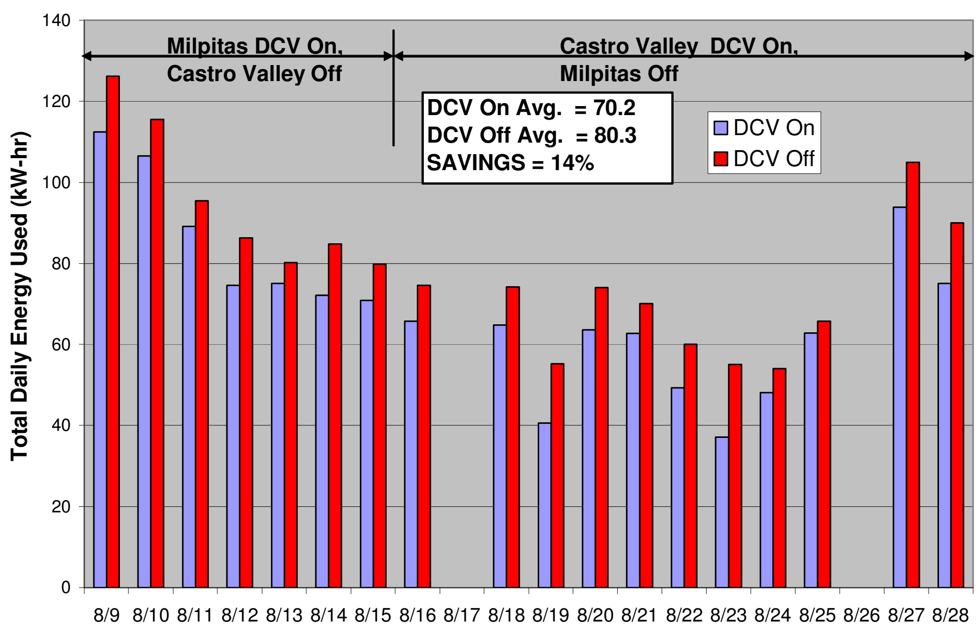


Figure 37. Cooling Energy Use for DCV On and Off at Bay Area McDonalds

Correlated Daily Energy Usage

Figure 38 shows daily energy usage for cooling as a function of daily average ambient temperature for the Milpitas site (bay area) for both DCV On and Off. The daily data correlates relatively well as a linear function of ambient temperature. For a hot day with an average temperature of 80º F, the estimated savings are about 12%. Figure 39 shows similar results for the other bay area site (Castro Valley). In this case, the savings are a little smaller than for the Milpitas site. This may be because this site has a greater occupancy, leading to higher ventilation rates for DCV On as compared with Milpitas.

Figure 40 shows daily energy usage for cooling as a function of daily average ambient temperature for the Bradshaw (Sacramento area) McDonalds for DCV On and Off. For a hot day with an average temperature of 80º F, the estimated savings are about 28%. These savings are considerably larger than those for the Bay area sites. For the same average daily temperature, the daytime temperatures are higher for Sacramento than the bay area leading to larger ventilation loads and greater savings with DCV. Also, the occupancy at the Bradshaw site appears to be lower than for the other McDonalds sites.

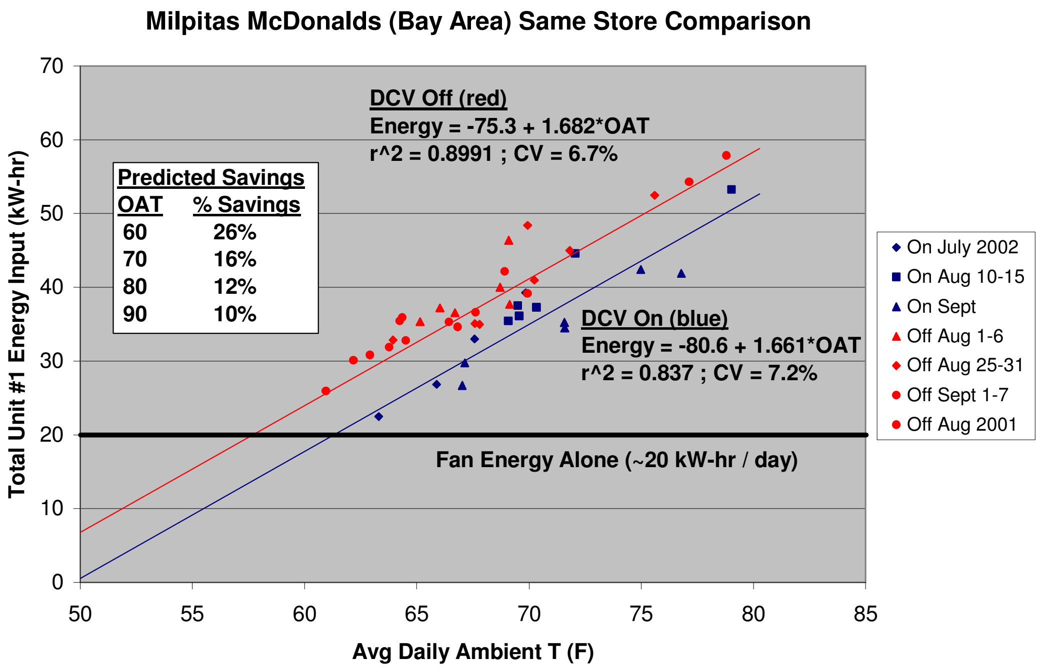


Figure 38. Correlated Daily Cooling Energy Use for DCV On and Off at Milpitas (Bay Area) McDonalds Site

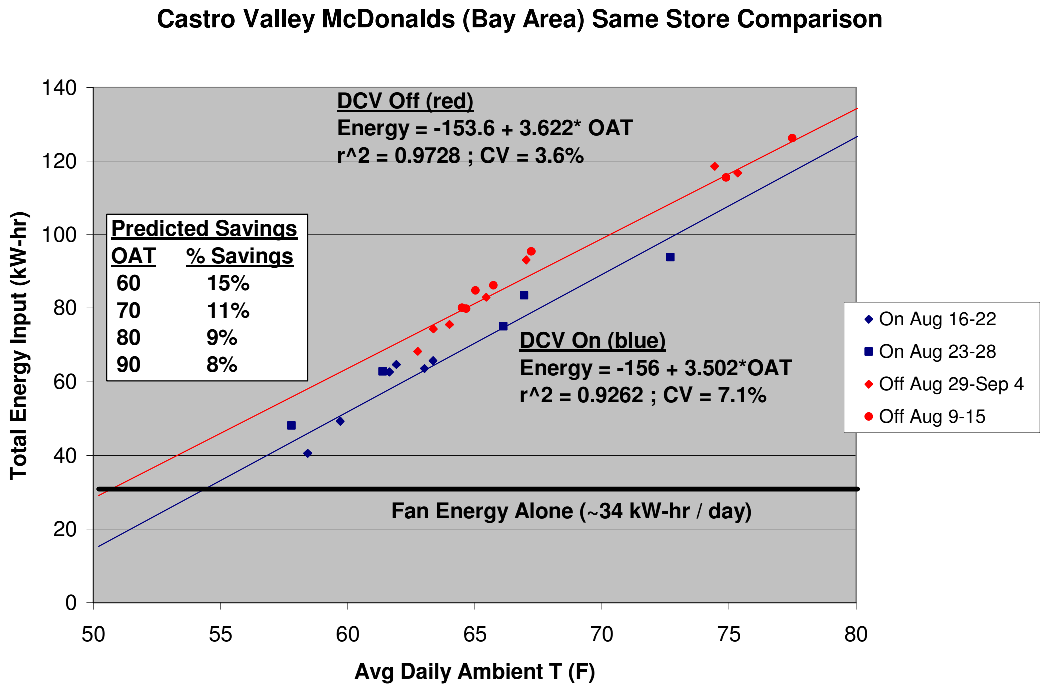


Figure 39. Correlated Daily Cooling Energy Use for DCV On and Off at Castro Valley (Bay Area) McDonalds Site

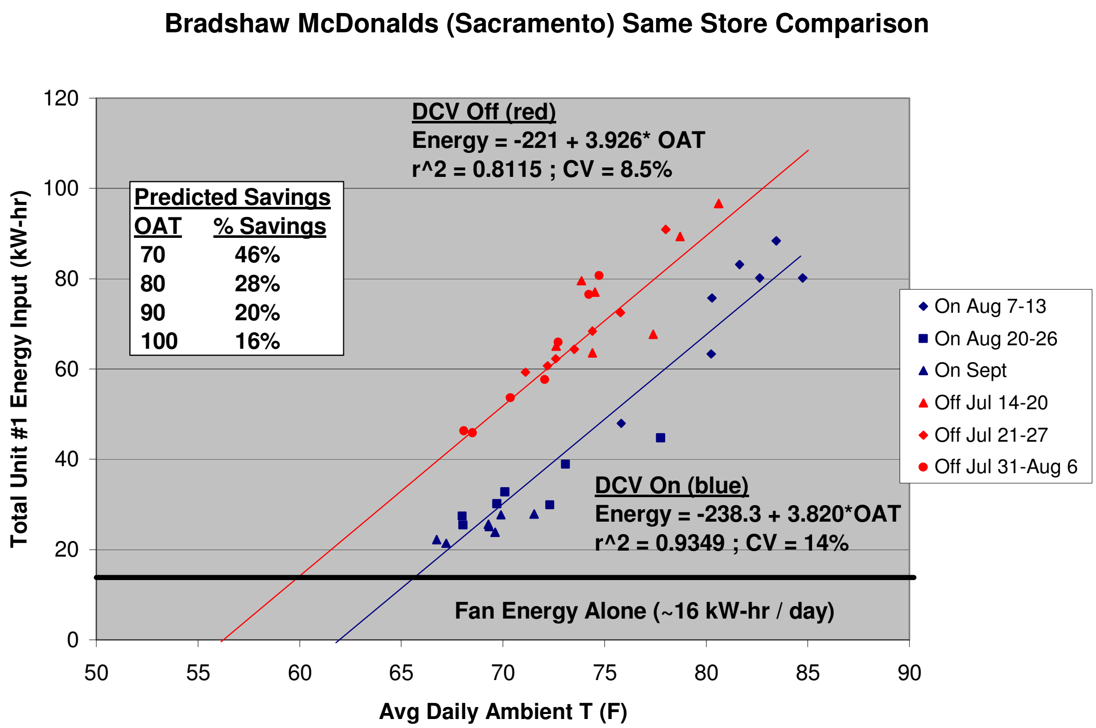


Figure 40. Correlated Daily Cooling Energy Use for DCV On and Off at Bradshaw (Sacramento) McDonalds Site

Table 13 summarizes the energy savings versus daily average ambient air temperature for the three McDonalds sites predicted from the time period with the available field data.

Table 13. Measured Savings Percentages with DCV On Control Strategy at McDonalds PlayPlace Areas

Daily Average Temperature (F)	Bradshaw Road (Sacramento)	Milpitas (Bay Area)	Castro Valley (Bay Area)
60	Not applicable (fan only)	26%	15%
70	46%	16%	11%
80	28%	12%	9%
90	20%	10%	8%

VSAT Comparisons

Site-specific VSAT models were prepared for the McDonalds sites and predicted daily energy consumption was compared with field measurements for same time periods used for Figure 38 to Figure 40. Parameters that describe the buildings and equipment were collected from site visits. An average occupancy profile was estimated from measurements of zone CO_2 concentrations and assumptions about average metabolic rates. Figure 41 to Figure 43 show that the predicted results generally match the measurements. The solid symbols represent the field measurements and the open symbols represent the VSAT predictions for the same dates and weather conditions. Regession correlation lines are also shown for the VSAT data. The field data and VSAT predictions are shown separately for DCV On and DCV Off operating modes for the Milpitas and Castro Valley sites for better clarity. At the Bradshaw site, there is enough separation between the DCV On and DCV Off data points to show them both on the same plot. On any given day, the model may not match the predictions very well due to differences in occupancy or other unmeasured differences. However, the correlations between daily energy usage and ambient temperature are close in most cases. In general, the estimated daily savings are smaller at lower average daily temperatures than for higher averages. On cooler days, economizer cooling is more significant and there is less potential for DCV savings. It was not possible to distinguish this trend from the experimental results due to the limited data and other confounding factors.

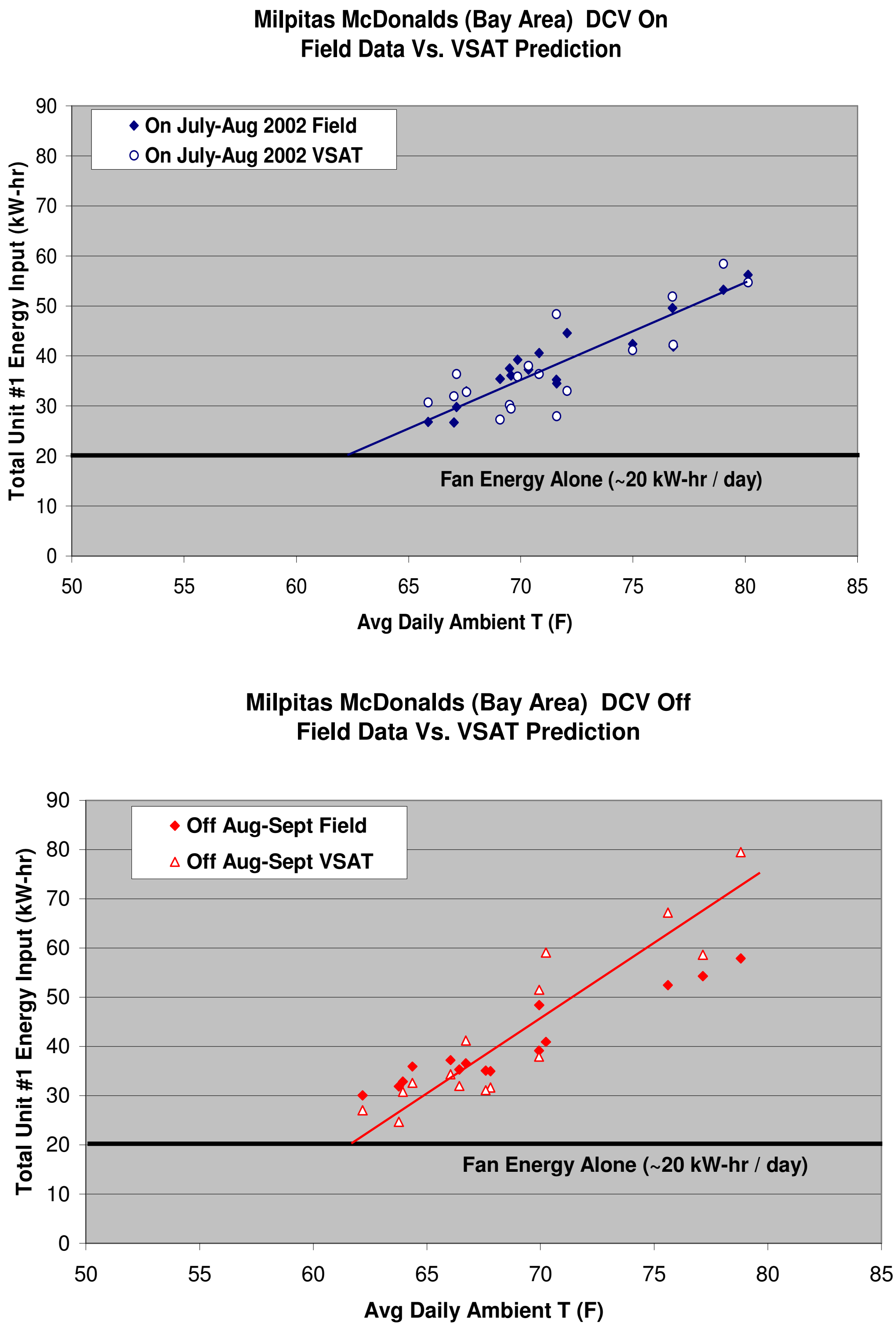
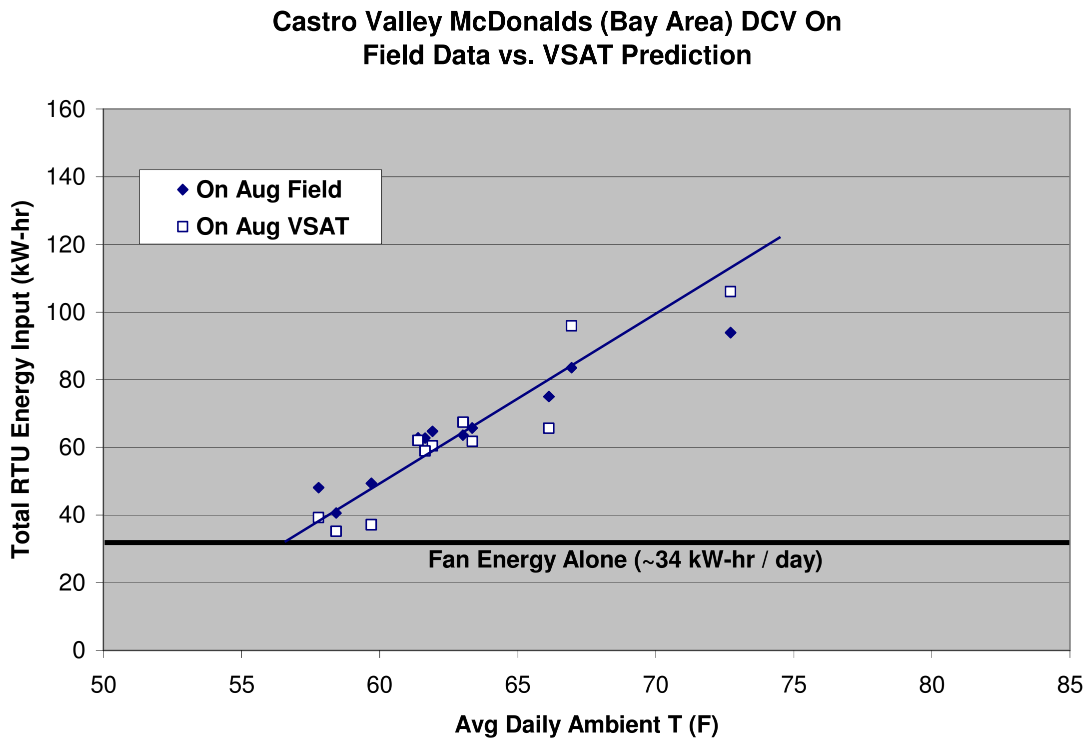


Figure 41. Comparison of Daily Cooling Energy Use at Milpitas (Bay Area) McDonalds Site



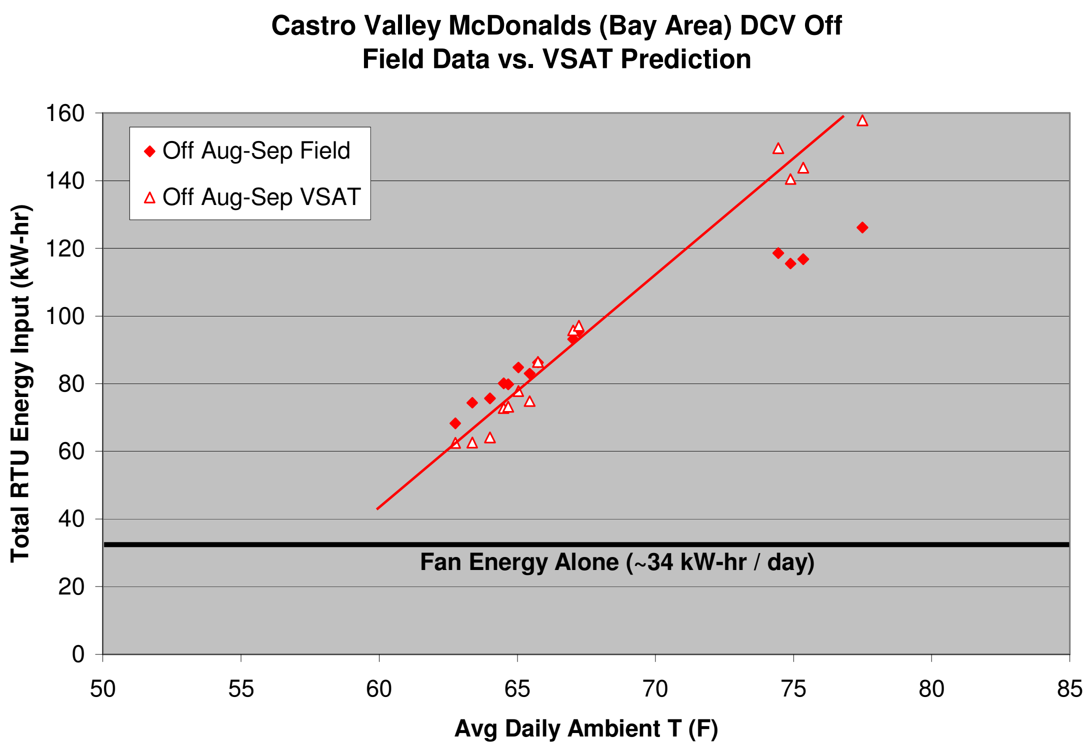


Figure 42. Comparison of Daily Cooling Energy Use at Castro Valley (Bay Area) McDonalds Site

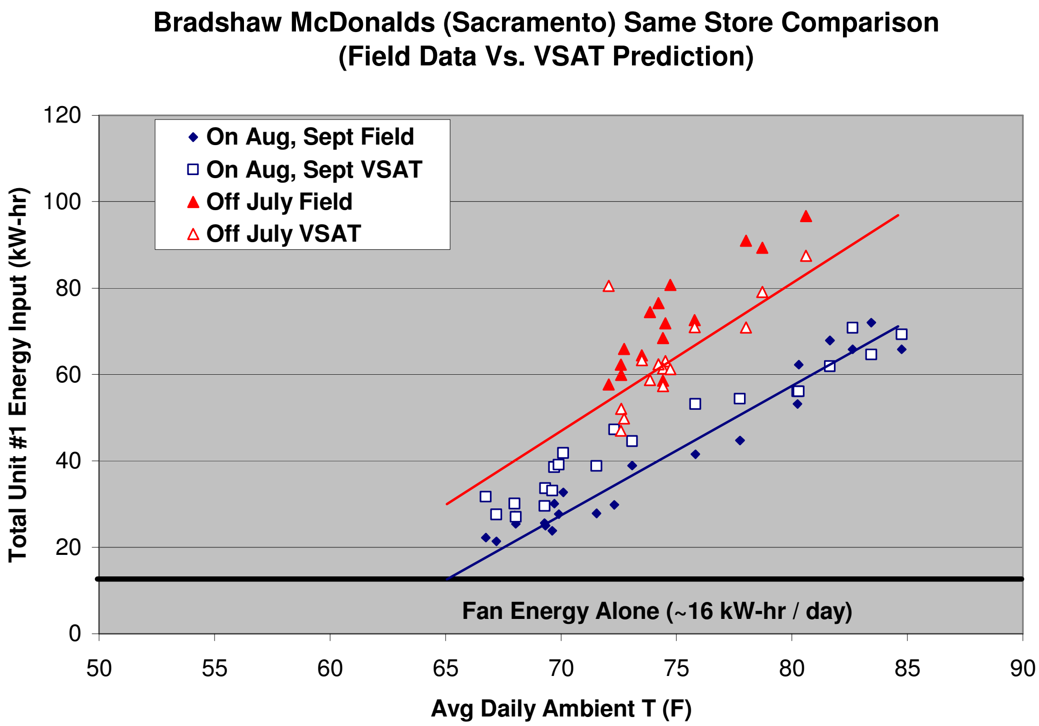


Figure 43. Comparison of Daily Cooling Energy Use at Bradshaw (Sacramento Area) McDonalds Site

Annual Cost Savings and Economic Analyses

The calibrated VSAT simulations for the field sites were used to evaluate annual operating cost savings, simple payback period, and return on investment for a 5-year period. The results are given in Table 14 and are based upon cooling season results only. Some additional cost savings will be realized from the heating season leading to lower payback periods. The payback periods are similar to those determined for the prototypical restaurant considered in the simulation section. Furthermore, the payback period is lower for the inland climate (Bradshaw) than for the coastal climate (Milpitas and Castro Valley). Note that the energy cost savings for the two restaurants in the coastal climate are similar, only differing by about $30. However, the economic payback and return on investment are very different. The Castro Valley site has only one rooftop unit, compared to two at Milpitas, and therefore shows a faster payback period and better return on the smaller initial investment. The field site comparison data in Figure 38 and Figure 39 were for a sampling of the entire cooling season when data were available. From the field site data alone during this sample period, the Milpitas site appears to have a slightly beter savings potential with DCV. However, when looked at on an annual basis and considering the initial cost of equipment for DCV, the Castro Valley site would be a better return on investment.

The rate of return is the interest rate that would provide an equivalent return on an investment; in this case the investment decision is whether to invest in a DCV system. The analysis is based on five years since that is the period for which the calibration of the CO_2 sensors are guaranteed. Many business may balk at considering investing in capital projects with a 3 or more year payback period, but the rate of return expected for the $900 per rooftop unit over the five year period is impressive for both the Bradsahw and Castro Valley site. A DCV retrofit would not make much economic sense for the Milpitas site. The assumed cost per rooftop unit is $900, as mentioned earlier. This cost is based on assuming the CO_2 sensors are located near the rooftop unit, such as in the return air stream. If the sensors were to be located in the occupied space, an additional cost would occur for running the wiring from the zone up to the rooftop unit.

Table 14. Predicted Cooling Season Savings with DCV On Control Strategy at McDonalds PlayPlace Areas from Calibrated VSAT Simulations

	Bradshaw Road (Sacramento)	Milpitas (Bay Area)	Castro Valley (Bay Area)
Compressor cooling savings (kW-hr)	1581	-4	228
Peak demand savings (kW)	6.1	2.3	2.0
Annual electrical energy cost savings ($)	$503	$277	$312
# RTU's per site	2	2	1
Total initial capital cost for DCV	$1,800	$1,800	$900
Simple payback period (years)	3.6	6.5	2.9
5 Year rate of return on investment in a DCV retrofit (%)	18.8%	-11.2%	34.8%

Indoor CO_2 Concentrations

Table 15 shows comparisons of average return air CO_2 concentrations during occupied periods for DCV On and DCV Off during the 2002 cooling season. The use of DCV results in higher CO_2 concentration levels for these test sites due to lower ventilation rates. This is consistant with the energy savings for DCV at these sites. The largest differences in CO_2 concentrations occur at the Bradshaw McDonalds. Recall that this site also had the largest energy savings for DCV. The Bradshaw site has lower average CO_2 concentrations for DCV Off than the other sites, implying that the occupancy is lower at this location. Lower occupancies relative to design occupancies generally lead to larger energy savings for DCV.

Table 15. Mean CO_2 Levels with DCV On and DCV Off Control Strategies at McDonalds PlayPlace Areas

DCV Control Strategy	Bradshaw Road (Sacramento)	Milpitas (Bay Area)	Castro Valley (Bay Area)
Off	496 ppm	541 ppm	572 ppm
On	575 ppm	613 ppm	615 ppm

Figure 44 through Figure 46 are histograms of the occupied hours that CO_2 concentrations fell within different bands for the Milpitas, Bradshaw, and Castro Valley sites. At the Milpitas and Bradshaw sites, the DCV controller was generally able to keep the return air CO_2 concentration at or below the 800 ppm set point. However, at the Castro Valley site, about 5% of the occupied hours were at CO_2 levels above 900 ppm.

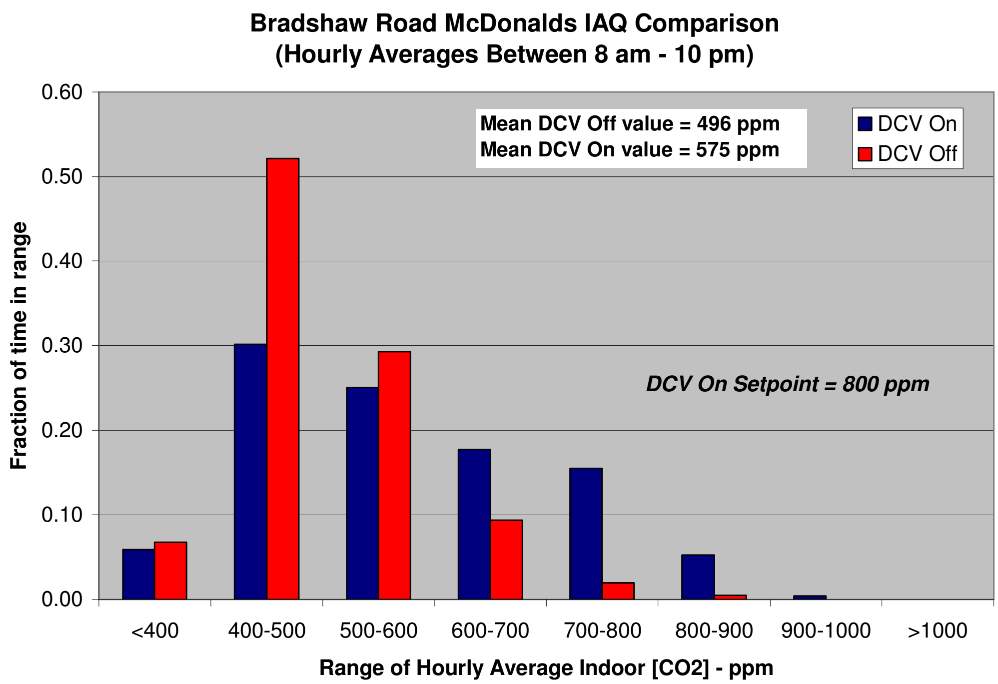


Figure 44. Histogram of Return Air CO_2 Concentrations at Bradshaw (Sacramento) McDonalds PlacePlace for DCV On and Off

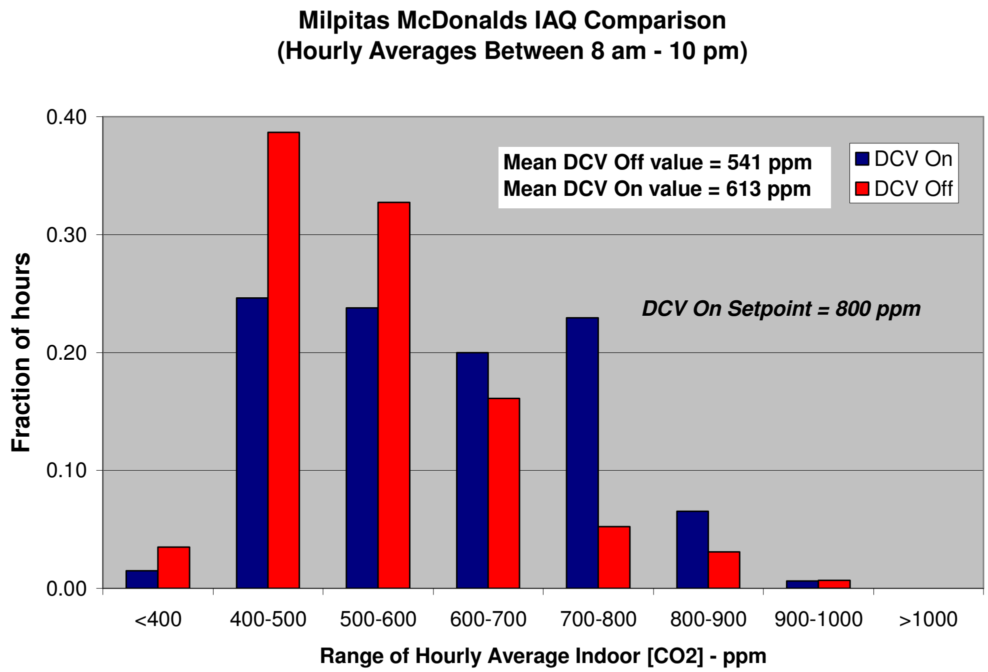


Figure 45. Histogram of Return Air CO_2 Concentrations at Milpitas (Bay Area) McDonalds PlacePlace for DCV On and Off

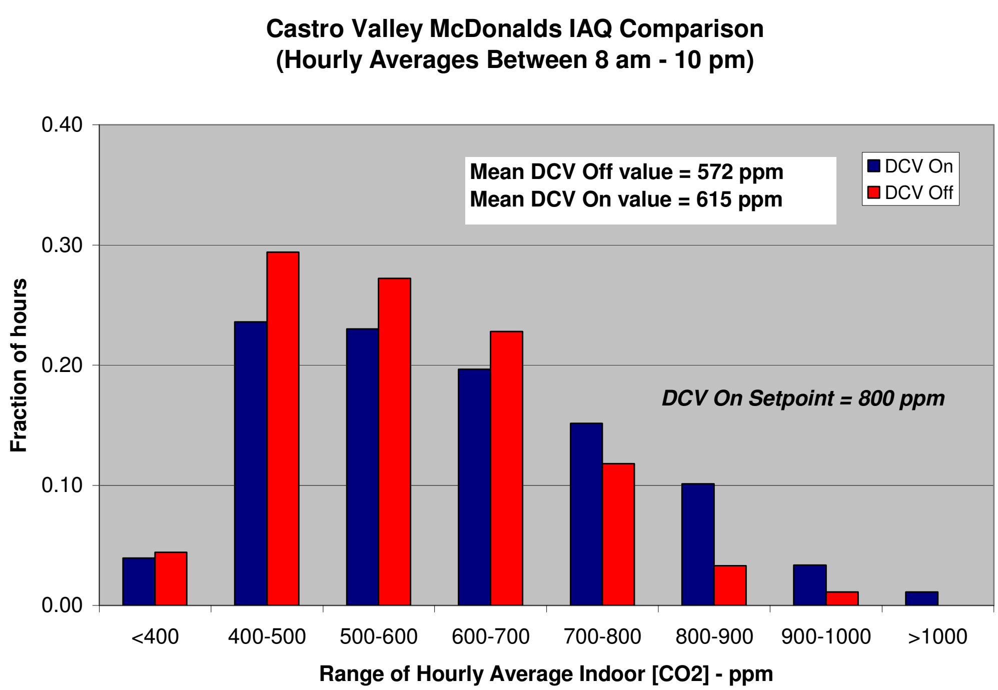


Figure 46. Histogram of Return Air CO_2 Concentrations at Castro Valley (Bay Area) McDonalds PlacePlace for DCV On and Off

Field Results for Modular Schools

Correlated Daily Energy Usage

Figure 47 and Figure 48 show daily energy usage for cooling as a function of daily average ambient temperature for the the Woodland site (Sacramento area) for both DCV On and Off. The average daily cooling energy use is nearly a linear function of ambient temperature. However, there appears to be no real difference in energy usage regardless of the control strategy chosen for the Woodland site. The average damper position for DCV On is essentially the same for both strategies implying that the rooms are fully occupied most of the time when the HVAC system is on and design ventilation air is required to maintain the CO_2 set point for DCV On. These schoolrooms are controlled by programmable thermostats that come on shortly before occupancy and turn off right as school lets out. Therefore, the rooms are most always occupied while the systems are on, which limits the potential for savings with DCV.

For the Woodland Gibson room 1, a special test was performed with the outdoor air damper set to match the amount of ventilation air provided with a unit that has a standard factory issue fixed louver configuration. The amount of ventilation air for this configuration is too small for the occupancy and is approximately 110 cfm or around 3 to 4 cfm per person. Therefore, typical installations for modular schoolrooms probably do not provide adequate indoor air quality. At this lower ventilation air flowrate, the energy usage was nearly the same for both DCV On or DCV Off.

Figure 49 and Figure 50 give similar results for the Oakland schoolrooms. The data do not correlate nearly as well with daily ambient temperature as for the other sites. Although it appears that DCV results in some energy savings, the differences are within the uncertainty of the correlation with ambient temperature.

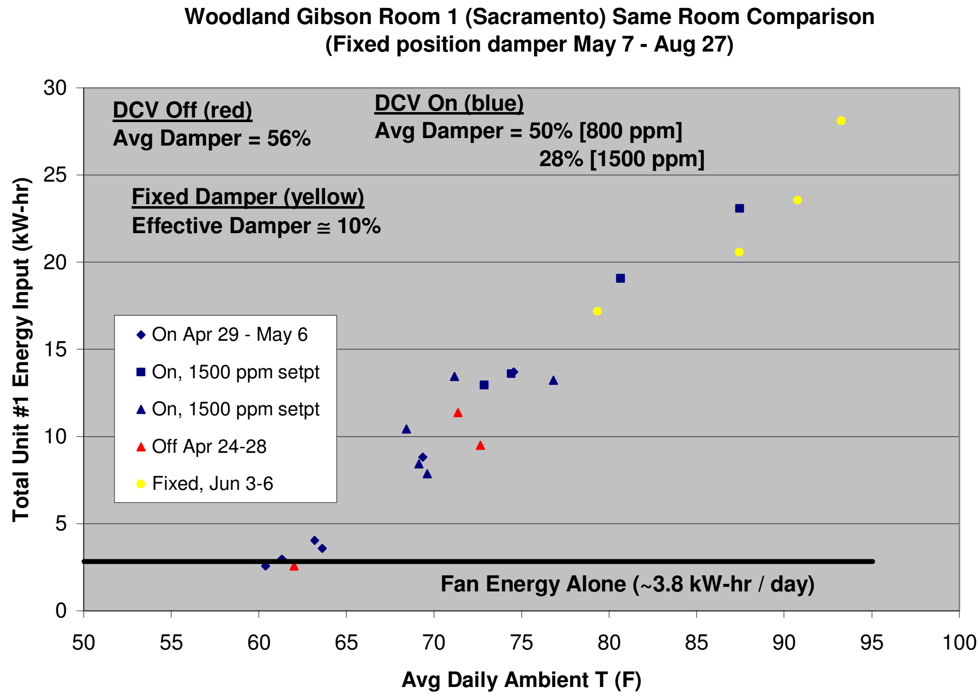


Figure 47. Correlated Daily Cooling Energy Use for DCV On and Off at Woodland (Sacramento) Schoolroom 1

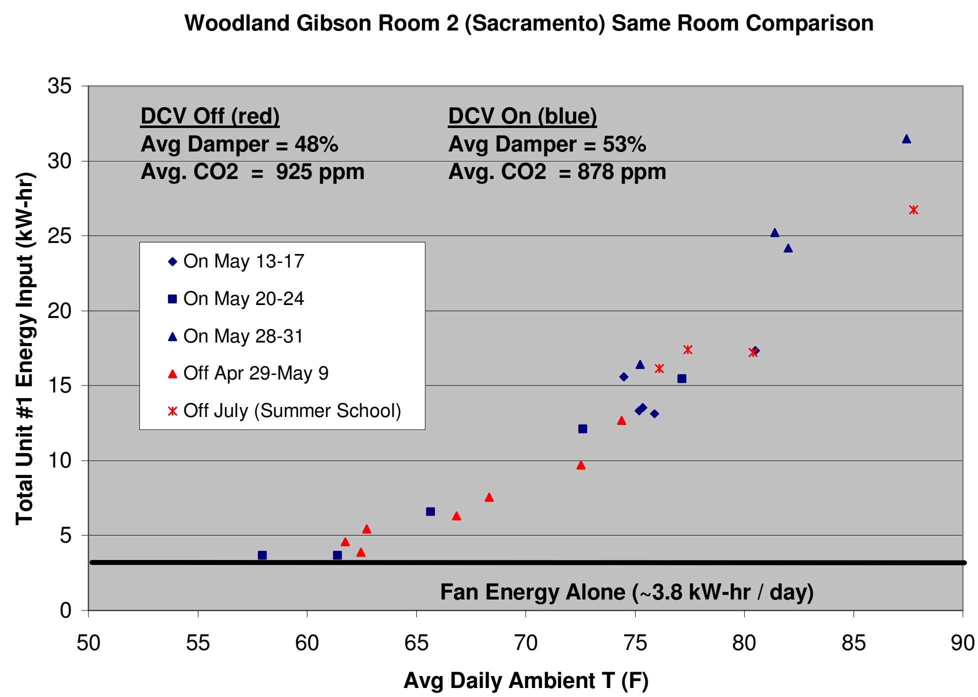


Figure 48. Correlated Daily Cooling Energy Use for DCV On and Off at Woodland (Sacramento) Schoolroom 2

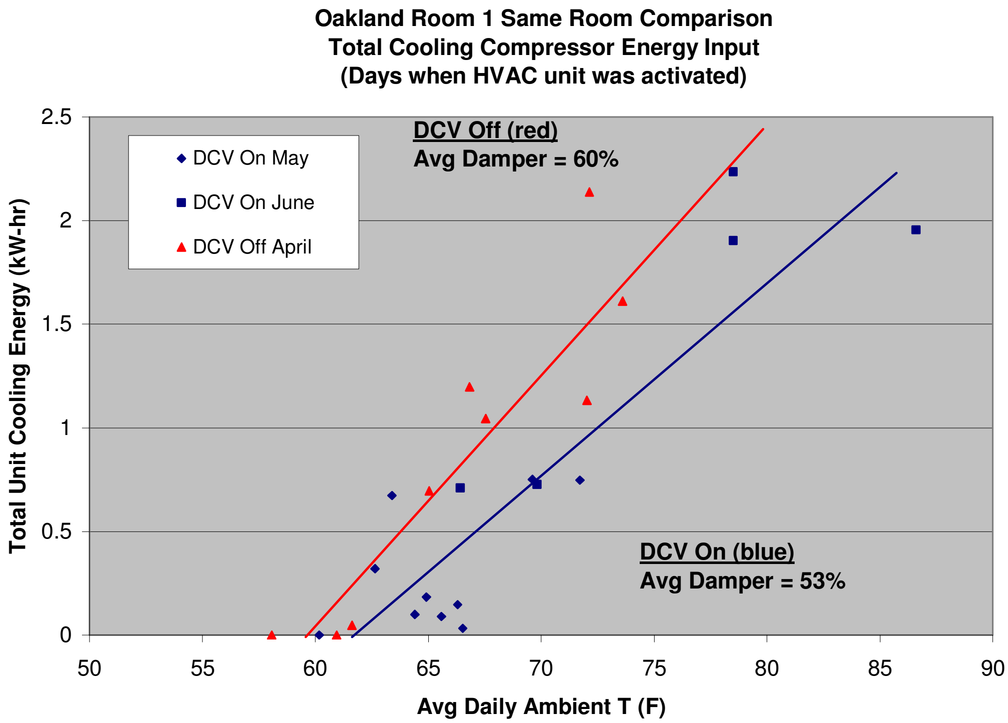


Figure 49. Correlated Daily Cooling Energy Use for DCV On and Off at Oakland Schoolroom 1

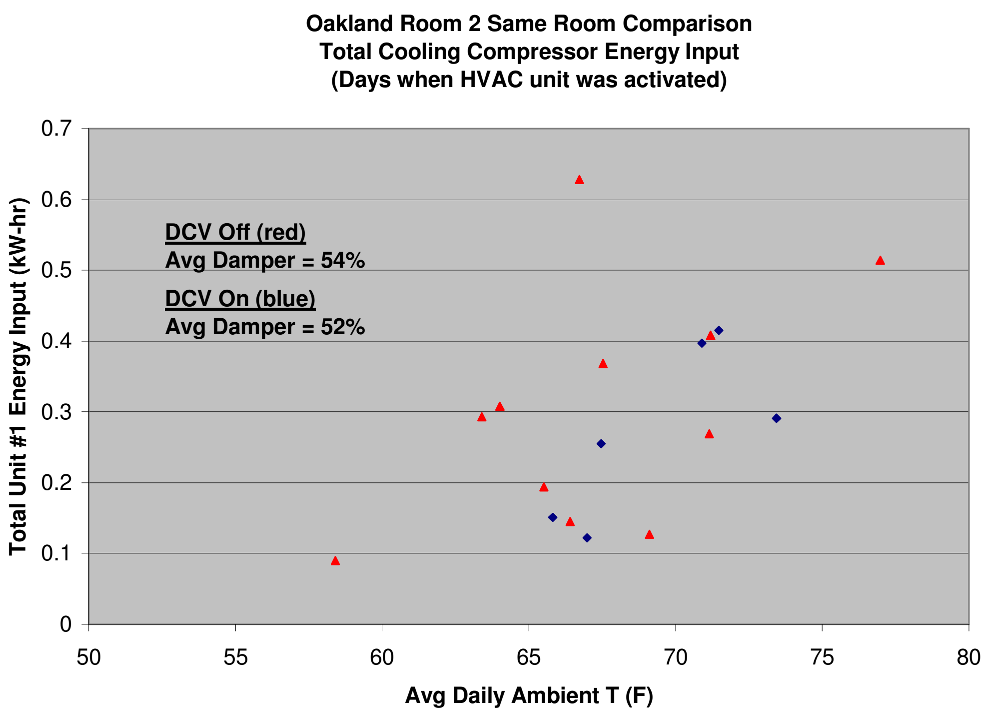


Figure 50. Correlated Daily Cooling Energy Use for DCV On and Off at Oakland Schoolroom 2

Indoor CO_2 Concentrations

Figure 51 is a histogram for return air CO_2 levels at one of the Gibson schoolrooms. Results are included for DCV On, DCV Off with fixed ventilation satisfying ASHRAE Standard 62-1999, and DCV Off with the ventilation airflow at the same level measured at a similar room that has only fixed air inlet louvers. Fixed air inlet louvers are the standard factory configuration for the sidewall mounted HVAC units, unless the economizer option is purchased with a modulating outdoor air damper. Since this is an additional option to the HVAC package, it is probably not installed in most school rooms.

The results in Figure 51 imply that the use of DCV results in better indoor air quality than for fixed ventilation determined according to ASHRAE Standard 62-1999. Possibly the metabolic rates assumed for application of the standard are lower than actually occur for this application. Furthermore, the use of the "Factory Standard" installation results in very high CO_2 concentrations. Over 60% of the occupied hours with the Factory Standard configuration had CO_2 levels that exceeded 1200 ppm. These levels violate California Title 24 requirements.

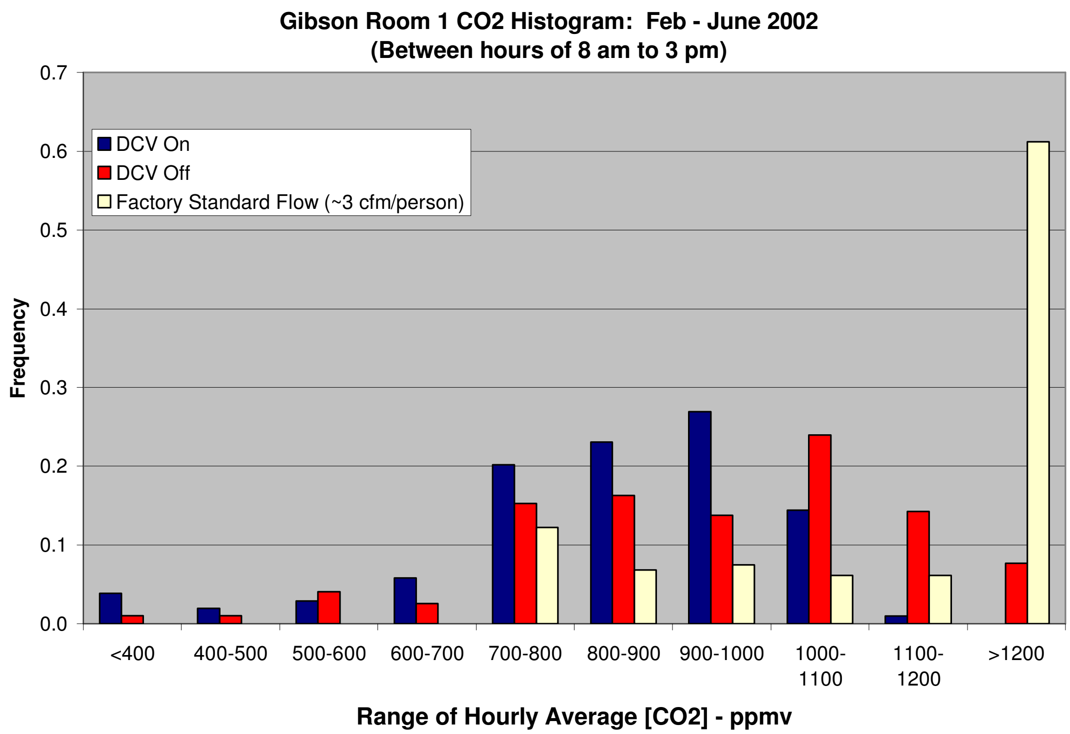


Figure 51. Histogram of Return Air CO_2 Concentrations at Woodland (Sacramento) Gibson Schoolroom 1 for DCV On, DCV Off, and Original Factory Installation

Figure 52 gives a histogram for the second Gibson schoolroom. Compared to room 1, the CO_2 levels are much higher for this room, implying a higher occupancy. However, there is a large number of hours for CO_2 concentrations above 1200 ppm with DCV Off that can't be explained by higher occupancy. This result may be due to problems with the controller. In some of the field sites, the minimum position for the outdoor air damper changes randomly at times and is not always maintained at the 40% set point for DCV Off.

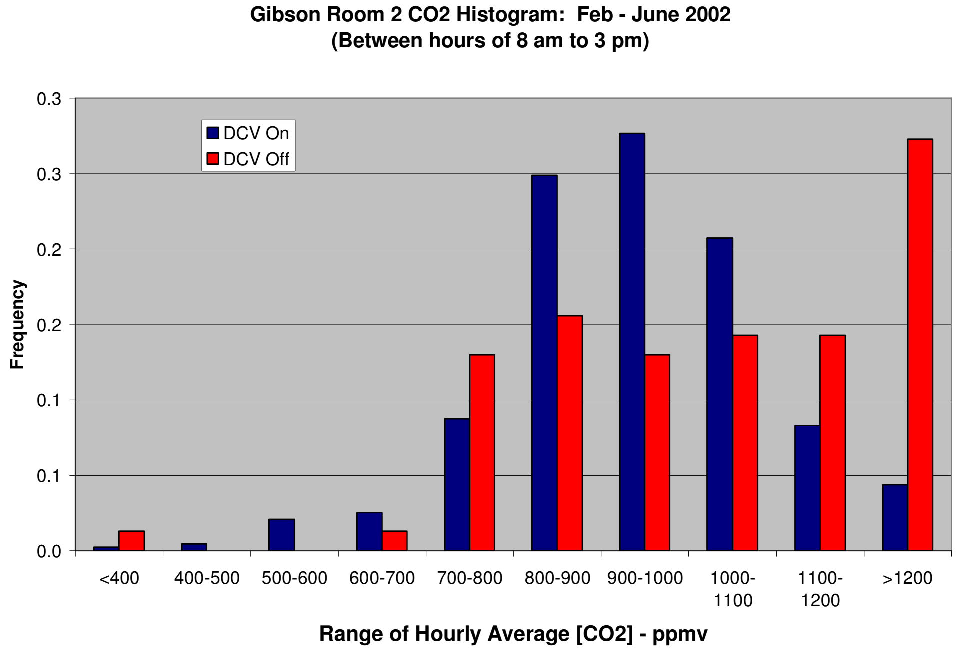


Figure 52. Histogram of Return Air CO_2 Concentrations at Woodland (Sacramento) Gibson Schoolroom 2 for DCV On and DCV Off

Field Results for Walgreens

Insufficient data are currently available to allow direct comparison of the DCV energy usage for cooling at the Walgreen sites. However, limited data for the late fall of 2002 at the Rialto site were used to validate a site-specific VSAT model. Figure 52 shows comparisons between daily measured and predicted energy usage for this site. The predictions of daily energy usage tend to be lower than the actual measurements for both DCV On and Off. However, the trends with respect to ambient temperature are similar. The measured performance is probably poorer than the predictions due to poor maintenance of the equipment at this site. The simulation could be improved through calibration of the equipment models. Figure 52 doesn't demonstrate significant savings for DCV. However, that is because the data are at low daily average ambient temperatures where an economizer operates a signficant portion of the time.

The VSAT simulation model was then used to predict total annual energy savings with a DCV retrofit for the Rialto Walgreens site. This comparison is given in Table 13. The comparison is only for the main retail store area and does not include the separate rooftop unit servicing the pharamacy area. These sites use heat pumps and thus electricity is the only energy source. The economic analysis for the Walgreen site does indeed provide an impressive case for installing DCV for a retail store in this climate. These results are very consistant with simulation results determined for the prototypical retail store in this climate zone. Actual cost savings realized depend on the assumption that the base case utilizes ventilation air flow rates that conform to the ASHRAE standard. For a retrofit installation, the economic benefit analysis also assumes that controllable air dampers, such as provided with an economizer system, are already installed. This was not the case for both the Walgreens and modular school sites which had to be modified for controllable air dampers as part of this study.

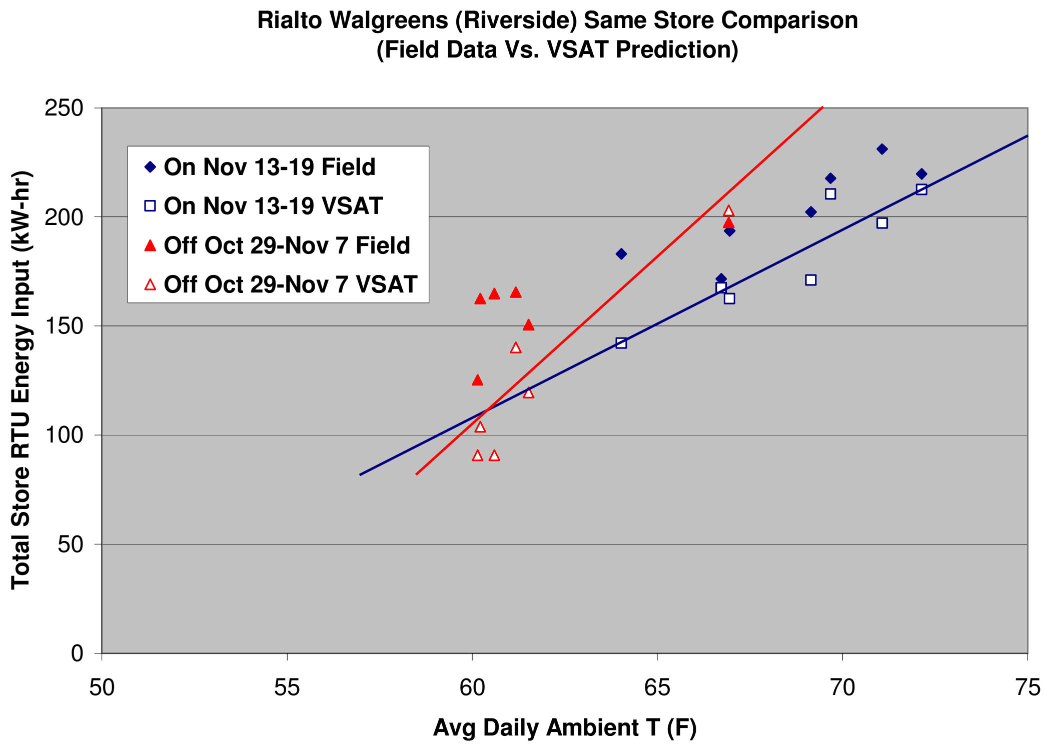


Figure 53. Comparison of Daily Cooling Energy Use at Rialto (LA Area, Inland Climate) Walgreens Site

Table 16. Predicted Annual Energy Savings with DCV On Control Strategy at a Walgreens Site Using VSAT Simulations

	Rialto (Riverside, CA)
Compressor power savings for heating and cooling (kW-hr)	16,391
Peak demand savings (kW)	17.1
Annual electrical energy cost savings ($)	$4,599
# RTU's per site	4
Total initial capital cost for DCV	$3,600
Simple payback period (years)	0.8
5 Year rate of return on investment in a DCV retrofit (%)	>300%

V. HPHR FIELD TESTING

A single field site was established for the heat pump heat recovery unit in order to verify that the equipment operates properly and that field performance is comparable to data obtained from the manufacturer and laboratory tests.

The heat pump was installed at Douglas Elementary School in Woodland, CA, in combination with a Carrier® 6-ton rooftop unit. Air inlet and outlet temperatures were measured using thermistors. Polymer capacitance humidity sensors were used to measure relative humidity at the inlets and outlets of the evaporator. Power consumption of the heat pump was monitored using a direct measure of the supply voltage and current draw from the unit. Two independent current measurements were taken in order to obtain both total and compressor power consumption. A more detailed description of the field site installation and setup is given by Braun and Mercer (2003b).

Figure 54 and Figure 55 show example operating conditions for cooling. Ambient air temperatures were very moderate throughout much of the day on August 2 and the heat pump did not operate very much during the first 4 occupied hours. The fan operated continuously for the entire occupied time to maintain proper ventilation. It's important to note that the fan power is very significant compared to the compressor power. The zone cooling set point for this day was approximately 72 F. The heat pump only operated to precondition the outside air for approximately one hour during the entire 8 hours of occupied cooling mode. Under these conditions, a system having an economizer with no energy recovery would have been would have used less energy and cost less to operate than the system with a heat pump. This is true throughout much of the cooling season in Woodland.

Figure 56 and Figure 57 give temperature and power measurements, respectively, for a much hotter day in Woodland. Ambient temperatures during occupied mode on July 24 were higher when compared to most other days in the data set from 2001 – 2002. For ambient temperatures between 90 F and 105 F, relative humidity varied from 24% to 4%, respectively. Therefore, even though ambient dry bulb temperatures were high, the actual wet bulb temperatures remained moderately low (~ 64 F) throughout the day. The heat pump operated several more hours on July 24 when compared to August 2 because of the higher building load, partly due to the higher ambient temperatures. For cooling mode, ambient wet bulb temperatures must exceed about 75 F and the heat pump must operate for a significant number of hours to enable a overall energy savings (see Braun and Mercer, 2003c).

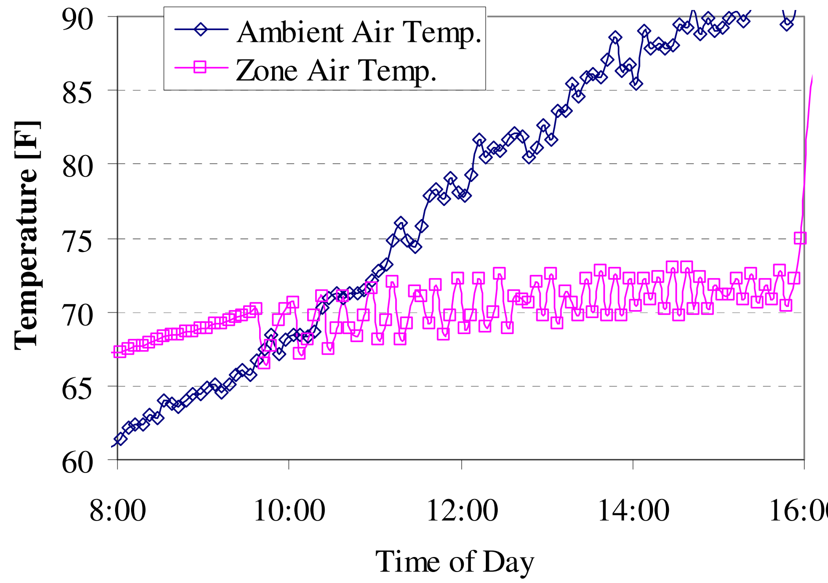


Figure 54. Occupied Daily Temperatures, August 2, 2002

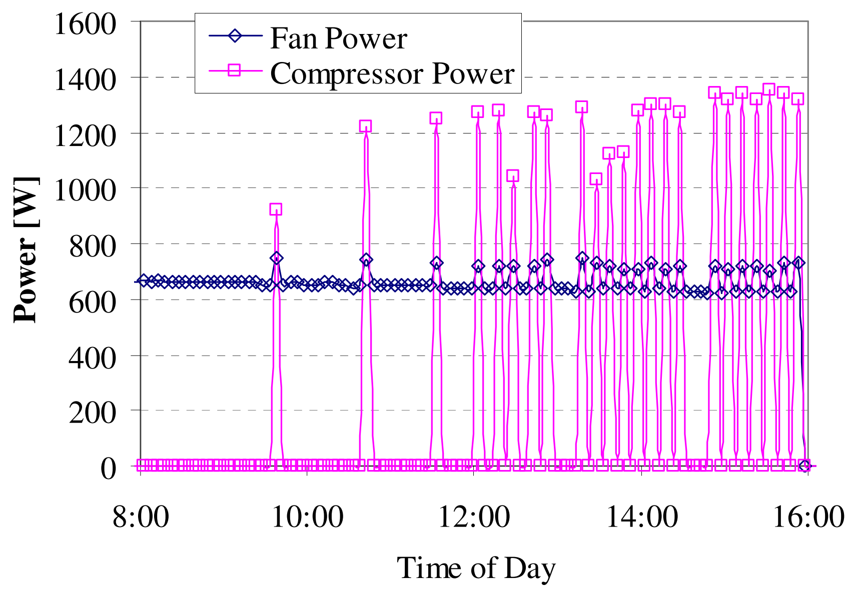


Figure 55. Occupied Daily Power, August 2, 2002

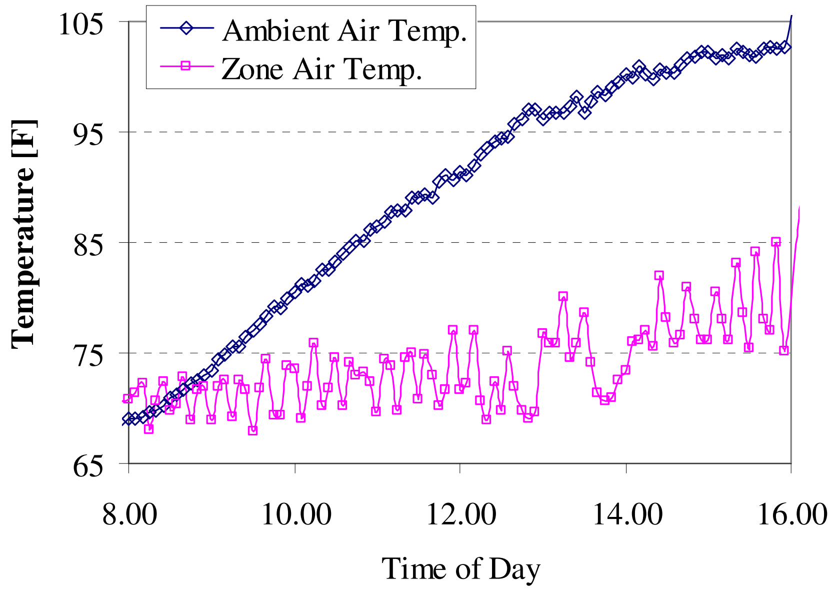


Figure 56. Occupied Daily Temperatures, July 24, 2002

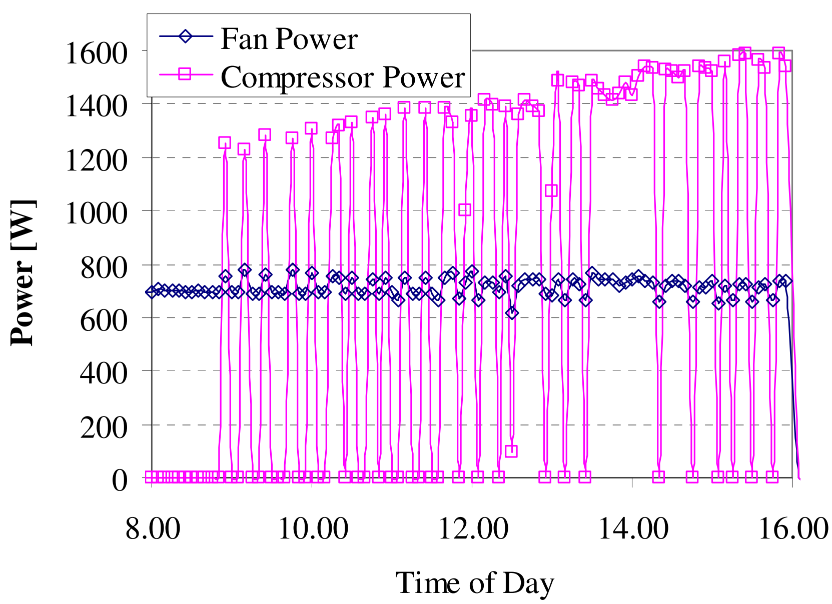


Figure 57. Occupied Daily Power, July 24, 2002

Steady-state operation of the heat pump occurred between 3:15 and 3:45 PM (7 – five-minute increment data points) on July 24. Figure 58 and Figure 59 show capacity and compressor power consumption for these steady-state points compared to model predictions, respectively. At steady-state conditions, the performance of the heat pump in the field is very close to the performance determined in the laboratory and published by the manufacturer. Furthermore, the model implemented within VSAT for the heat pump accurately predicts capacity and compressor power when compared to recorded field data for steady-state conditions. However, the VSAT model does not include energy losses due to on/off cycling. Therefore, the VSAT predictions tend to be optimistic with respect to energy savings associated with the heat pump heat recovery unit.

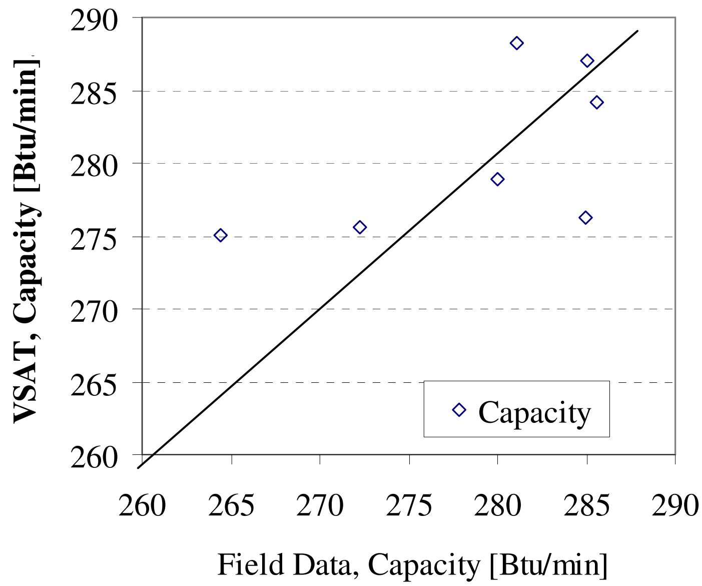


Figure 58. Predicted vs. Recorded Capacity (July 24, 2002)

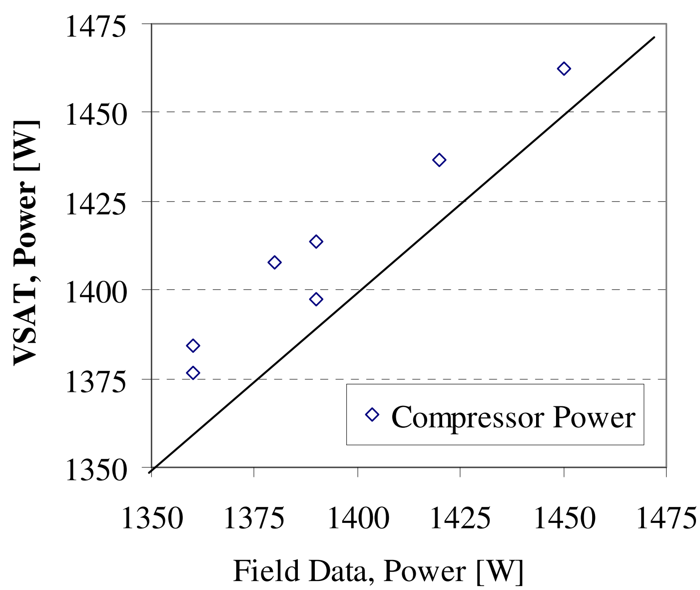


Figure 59. Predicted vs. Recorded Compressor Power (July 24, 2002)

Figure 60 and Figure 61 show example conditions for a day during the heating season in Woodland. The ambient temperature was near freezing early in the morning, but steadily increased up to 55 F by the end of the occupied time. The zone heating set point for this day was approximately 65 F. However, as in cooling season, the zone temperature set points were frequently altered.

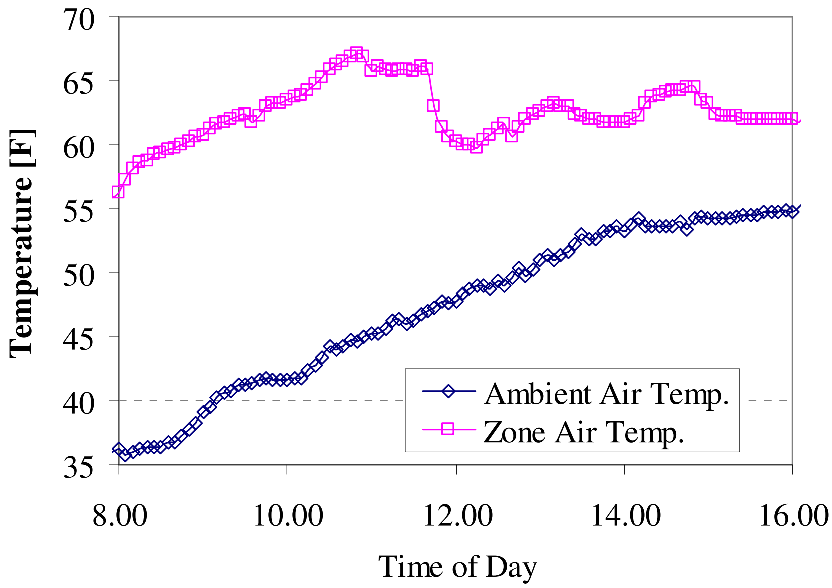


Figure 60. Occupied Daily Temperatures, Jan. 17, 2002

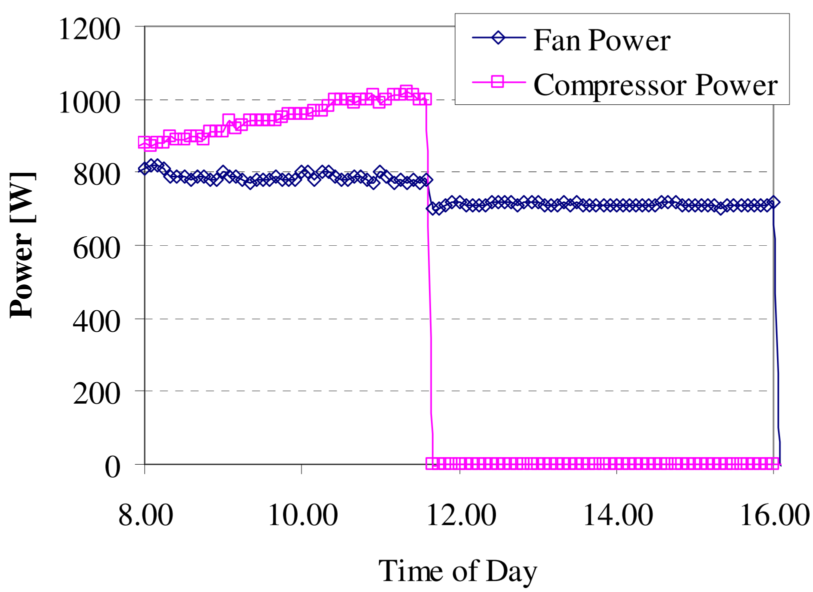


Figure 61. Occupied Daily Power, Jan. 17, 2002

Steady-state operation of the heat pump occurred between 8:00 and approximately 11:30 AM on January 17. For this 3 ½ hour time period, heat pump compressor power increased as ambient temperature increased. A total of 40, five-minute increment steady-state data points were used for comparisons with the VSAT model. Figure 62 and Figure 63 show capacity and compressor power consumption for these steady-state points compared to model predictions, respectively. For steady-state operation, the heat pump component model within VSAT accurately predicts capacity and compressor power compared to recorded field data.

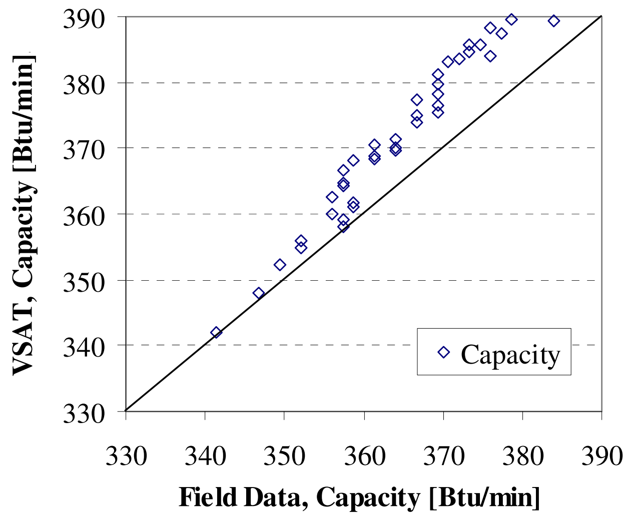


Figure 62. Predicted vs. Recorded Capacity (Jan. 17, 2002)

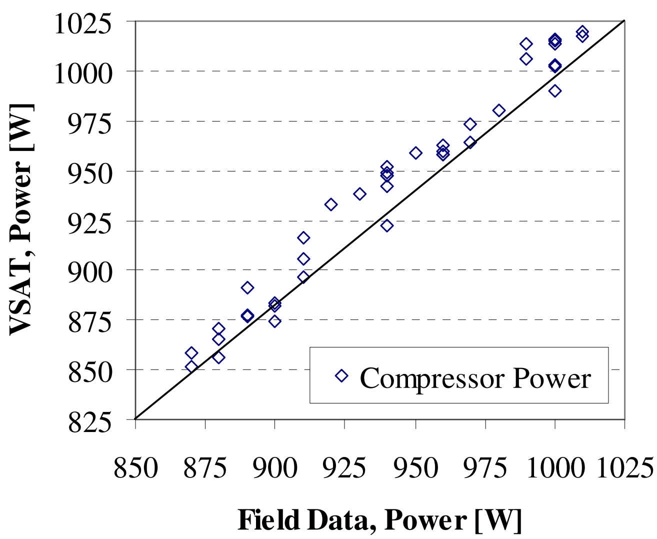


Figure 63. Predicted vs. Recorded Compressor Power (Jan. 17, 2002)

VI. CONCLUSIONS AND RECOMMENDATIONS

Demand-controlled ventilation coupled with an economizer (DCV+EC) was found to give the largest cost savings relative to an economizer only system for a number of different prototypical buildings and systems evaluated in the 16 California climate zones. These results were independent whether DCV is considered for retrofit or new applications. DCV reduces ventilation requirements and loads whenever the economizer is not enabled and the occupancy is less than the peak design value typically used to establish fixed ventilation rates according to ASHRAE Standard 62-1999. Lower ventilation loads lead to lower equipment loads, energy usage and peak electrical demand. The greatest cost savings occur for buildings that have low average occupancy relative to their peak occupancy, such as auditoriums, gyms and retail stores. From a climate perspective, the greatest savings and lowest payback periods occur in extreme climates (either hot or cold). The mild coastal climates have smaller savings and longer payback periods. In most cases, the payback period associated with DCV+EC was less than 2 years.

The heat pump heat recovery (HPHR) system did not provide positive cost savings for many situations investigated for California climates. Heating requirements are relatively low for California climates and therefore overall savings are dictated by cooling season performance. The cooling COP of the HPHR system must be high enough to overcome additional cycling losses from the primary air conditioner compressor, additional fan power associated with the exhaust and/or ventilation fan, additional cooling requirements due to a higher latent removal and a lower operating COP for the primary air conditioner compressor because of a colder mixed air temperature. In addition, the HPHR system is an alternative to an economizer and so economizer savings are also lost when utilizing this system. There are not sufficient hours of ambient temperatures above the breakeven points to yield overall positive savings with the HPHR system compared to a base case system with an economizer for the prototypical buildings in California climates.

The breakeven ambient temperatures for positive savings with the HXHR system are much lower than for the HPHR system because the energy recovery (and reduced ventilation load) does not require additional compressor power. The primary penalty is associated with increased fan power due to an additional exhaust fan. In addition, as with the HPHR system, the HXHR system is an alternative to an economizer. Therefore, economizer savings are also lost when utilizing this system. Although positive savings were realized for a number of different buildings and climate zones, the HXHR system had greater operating costs than the DCV system for all cases considered. Furthermore, the initial cost for an HXHR system is higher than a DCV system and also requires higher maintenance costs. Payback for the enthalpy exchanger was found to be greater than 7 years for most all areas of California, except for some building types in climate zone 15. However, paybacks were calculated assuming a retrofit application. The use of an enthalpy exchanger would lead to a smaller design load for the HVAC equipment which could impact the overall economics.

For humid climates (outside of California), the alternative ventilation strategies provide lower zone humidity levels than a conventional system during the cooling season. Typically, DCV provides the lowest zone humdities, followed by the HXHR system, and then the HPHR system.

The savings and trends determined through simulation for DCV were verified through field testing in a number of sites. Field sites were established for three different building types in two different climate zones within California. The building types are: 1) McDonalds PlayPlace® areas, 2) modular school rooms, and 3) Walgreens drug stores. In each case, nearly duplicate test buildings were identified in both coastal and inland climate areas. For cooling, greater energy and cost savings were achieved at the McDonalds PlayPlaces and Walgreens than for the modular schoolrooms. Primarily, this is because these buildings have more variability in their occupancy than the schoolrooms. The largest energy and cost savings were achieved at the Walgreens in Rialto, followed by the Bradshaw McDonalds PlayPlaces. The Rialto Walgreens appears to have the lowest occupancy and is located in a relatively hot climate with relatively large ventilation loads. The Bradshaw McDonalds PlacePlace appears to have the lowest average occupancy level compared to the other McDonalds PlacePlaces. This site is located in Sacramento and has larger ventilation and total cooling loads than the bay area McDonalds. The payback period for the Rialto Walgreens is less than a year and is between 3 and 6 years for the McDonalds PlayPlaces.

There were no substantial cooling season savings for the modular school rooms. The occupancy for the schools is relatively high with relatively small variability. The school sites are also on timers or controllable thermostats that mean the HVAC units only operate during the normal school day. The schools are also generally unoccupied during the heaviest load portion of the cooling season. Furthermore, the results imply that the average metabolic rate of the students may be higher than the value used in ASHRAE Standard 62-1999 to establish a fixed ventilation rate. In fact, the DCV control resulted in lower CO_2 concentrations than for fixed ventilation rate in the Woodland modular schoolrooms.

The field data confirmed that the steady-state performance of the heat pump in the field is very close to the performance determined in the laboratory and published by the manufacturer for both cooling and heating modes. Furthermore, the model implemented within VSAT for the heat pump accurately predicts capacity and compressor power when compared to recorded field data for steady-state conditions.

For most all locations throughout the state of California, demand-controlled ventilation with an economizer is the recommended ventilation strategy. An enthalpy exchanger is viable in many situations, but DCV was found to have better overall economics for retrofit applications. Heat pump heat recovery is not recommended for California. This technology would make more sense in cold climates where heating costs are more significant. The savings potential for all ventilation strategies is greater in cold climates where heating dominates.

VII. REFERENCES

ASHRAE (1999), *ANSI/ASHRAE Standard 62-1999: Ventilation for Acceptable Indoor Air Quality,* ASHRAE, 1791 Tullie Circle, NE, Atlanta, GA 30329.

Balcomb, J. D. (2002), *Mastering Energy-10: A User Manual for Version 1.5*, National Renewable Energy Laboratory, Golden, CO.

Brandemuehl, M.J. and Braun, James E. (2002), *The Savings Estimator,* Honeywell International Inc., Morristown, NJ., http://thermal.ies.lafayette.in.us/savest

Brandemuehl, M.J. and Braun, James E. (1999), *The Impact of Demand-Controlled and Economizer Ventilation Strategies of Energy Use in Buildings,* ASHRAE Transactions 105 (2): 39-50.

Brandemuehl, M.J., Gabel, S., and Andresen, I. (2000), *HVAC2 Toolkit: Algorithms and Subroutines for Secondary HVAC System Energy Calculations,* ASHRAE, 1791 Tullie Circle, NE, Atlanta, GA 30329.

Braun, J.E., and Mercer, K. (2002), *Laboratory Test Evaluation And Field Installation Of The Energy Recycler® Heat Pump as Deliverable 4.2.4a*, California Energy Commission, Sacramento, CA.

Braun, J.E. and Mercer, K. (2003), *VSAT – Ventilation Strategy Assessment Tool as Deliverables 3.1.2, 3.2.1 and 4.2.2,* California Energy Commission, Sacramento, CA.

Braun, J.E. and Mercer, K. (2003), *Field Data Analysis of the Energy Recycler® Heat Pump as Deliverable 4.2.5a*, California Energy Commission, Sacramento, CA.

Braun, J.E. and Mercer, K. (2003), *Operating Cost Assessments And Comparisons For Demand Controlled Ventilation, Heat Hump Heat Recovery And Enthalpy Exchangers as Deliverable 4.2.3a*, California Energy Commission, Sacramento, CA.

Carpenter, S.C. (1996), *Energy and IAQ Impacts of CO_2-Based Demand-Controlled Ventilation,* ASHRAE Transactions 102 (2): 80-88.

Donnini, G., Haghighat, F., and Hguyen, V. H. (1991), *Ventilation Control of Indoor Air Quality, Thermal Comfort, and Energy Conservation by CO_2 Measurement,* Proceedings of the 12th AIVC Conference Air Movement and Ventilation Control within Buildings: 311-331.

Emmerich, Steven J. and Persily, Andrew K. (2001), *State-of-the-Art Review of CO_2 Demand Controlled Ventilation Technology and Application,* Technical Report NISTIR 6729, National Institute of Standards and Technology, Gaithersburg, MD.

Fehrm, M., Reiners, W., and Ungemach, M. (2002), *Exhaust Air Heat Recovery in Buildings,* International Journal of Refrigeration 25: 439-449.

Gabel, S. D., Janssen, J., Christoffel, J., and Scarborough, S. (1986), *Carbon Dioxide Based Ventilation Control System Demonstration,* Technical Report DE-AC79-84BP15102, U.S. Department of Energy.

Haghighat, F., Zmeureanu, R., and Donnini G. (1993), *Energy Savings in a Building by a Demand Controlled Ventilation System,* Proceedings of the 6th International Conference on Indoor Air Quality and Climate. 5: 51-56.

Huang, Y.J., Akbari, H., Rainer, L., and Ritschard, R.L. (1990), *481 Prototypical Commercial Buildings for Twenty Urban Market Areas (Technical documentation of building loads data base developed for the GRI Cogeneration Market Assessment Model),* LBNL Report 29798, Lawrence Berkeley National Laboratory, Berkeley, CA.

Huang, Y.J., and Franconi, E. (1995), *Commercial Heating and Cooling Load Component Analysis,* LBNL Report 38970, Lawrence Berkeley National Laboratory, Berkeley, CA.

Janssen, J., Hill, T., Woods, J., and Maldonado, E. (1982), *Ventilation for Control of Indoor Air Quality: A Case Study,* Environment International. 8: 487-496.

Knoespel, P., Mitchell, J., and Beckman, W. (1991), *Macroscopic Model of Indoor Air Quality and Automatic Control of Ventilation Airflow,* ASHRAE Transactions 97 (2): 1020-1030.

Lawrence T.M. and Braun, J.E. (2003), *Initial Cooling And Heating Season Field Evaluations For Demand-Controlled Ventilation as Deliverable 3.1.3a and 3.1.4a*, California Energy Commission, Sacramento, CA.

Rengarajan, K., Shirey III, D., and Raustad, D. (1996), *Cost-Effective HVAC Technologies to Meet ASHRAE Standard 62-1989 in Hot and Humid Climates*, ASHRAE Transactions 102 (1): 166-182.

Shirey III, D., and Rengarajan, K. (1996), *Impact of ASHRAE Standard 62-1989 on Small Florida Offices,* ASHRAE Transactions 102 (1): 153-165.

Slayzak, S. J., Pesaran, A. A., and Hancock, C. E. (1998), *Experimental Evaluation of Commercial Desiccant Dehumidifier Wheels*, NREL Technical Report.

Slayzak, S. J., and Ryan, J. P. (1998), *Instrument Uncertainty Effect on Calculation of Absolute Humidity Using Dew Point, Wet Bulb, and Relative Humidity Sensors*, International Solar Energy Conference: 473-479.

Stiesch, G., Klein, S. A., and Mitchell, J. W. (1995), *Performance of Rotary Heat and Mass Exchangers*, International Journal HVAC&R Research 1(4): 308-323.

TRNSYS (2000), *TRNSYS Users Guide: A Transient Simulation Program*, Solar Energy Laboratory, University of Wisconsin-Madison. http://sel.me.wisc.edu/trnsys/Default.htm

Zamboni, M., Berchtold, O., Filleux, C., Fehlmann, J., and Drangsholt, F. (1991), *Demand Controlled Ventilation – An Application of Auditria,* Proceedings of the 12th AIVC Conference Air Movement and Ventilation Control within Buildings: 143-155.

APPENDIX A – PROTOTYPICAL BUILDING DESCRIPTIONS

Seven different types of buildings are considered in VSAT: small office, school class wing, retail store, restaurant dining area, school gymnasium, school library, and school auditorium. Descriptions for these buildings were obtained from prototypical building descriptions of commercial building prototypes developed by Lawrence Berkeley National Laboratory (Huang, et al., 1990 & Huang, et al., 1995). These reports served as the primary sources for prototypical building data. However, additional information was obtained from DOE-2 input files used by the researchers for their studies.

Tables A.1 through A.7 contain information on the geometry, construction materials, and internal gains used in modeling the different buildings. Although not given in these tables, the walls, roofs and floors include inside air and outside air thermal resistances. The window R-value includes the effects of the window construction and inside and outside air resistances. Table A.8 lists the properties of all construction materials and the air resistances. The geometry of each of the buildings is assumed to be rectangular with four sides and is specified with the following parameters: 1) floor area, 2) number of stories, 3) aspect ratio, 4) ratio of exterior perimeter to total perimeter, 5) wall height and 6) ratio of glass area to wall area. The aspect ratio is the ratio of the width to the length of the building. However, exterior perimeter and glass areas are assumed to be equally distributed on all sides of the building, giving equal exposure of exterior walls and windows to incident solar radiation. The four exterior walls face north, south, east, and west.

The user can specify occupancy schedules, but default values are based upon the original LBNL study. In the LBNL study, the occupancy was scaled relative to a daily average maximum occupancy density (people per 1000 ft^2). In VSAT, the user can specify a peak design occupancy density (people per 1000 ft^2) that is used for determining fixed ventilation requirements (no DCV). This same design occupancy density is used as the scaling factor for the hourly occupancy schedules. As a result, the original LBNL occupancy schedules were rescaled using the default peak design occupancy densities.

The heat gains and CO_2 generation per person depend upon the type of building (and associated activity). Design internal gains for lights and equipment also depend upon the building and are scaled according to specified average daily minimum and maximum gain fractions. For all of the buildings, the lights and equipment are at their average maximum values whenever the building is occupied and are at their average minimum values at all other times.

Zone thermostat set points can be set for both occupied and unoccupied periods. The default occupied set points for cooling and heating are 75 F and 70 F, respectively. The default unoccupied set points for cooling (setup) and heating (setback) are 85 F and 60 F, respectively. The lights are assumed to come on one hour before people arrive and stay on one hour after they leave. The occupied and unoccupied set points follow this same schedule.

Table A.1. Office Building Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.58	
Shading Coefficient	0.75	
Area ratio (window/wall)	0.15	
Exterior Wall Construction		
Layers	1" stone R-5.6 insulation R-0.89 airspace 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") 4" lightweight concrete R-12.6 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete Carpet and pad	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	6600	
Wall height, ft	11	
Internal mass, lb/ft^2	25	
Number of stories	1	
Aspect Ratio	0.67	
Ratio of exterior perimeter to floor perimeter	1.0	
Design equipment gains, W/ft^2	0.5	
Design light gains, W/ft^2	1.7	
Ave. daily min. lights/equip. gain fraction	0.2	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	250	
CO_2 people generation, L/min-person	0.33	
Design occupancy for vent., people/1000 ft^2	7	
Design ventilation, cfm/person	20	
Average weekday peak occucpancy, ft^2/person	470	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-7 8 9 10-16 17 18-24	Values 0.0 0.33 0.66 1.0 0.5 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-8 9 10-12 12-13 13-24	Values 0.0 0.15 0.2 0.15 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-12	Value 1.0

Table A.2. Restaurant Dining Area Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.53	
Shading Coefficient	0.8	
Area ratio (window/wall)	0.15	
Exterior Wall Construction		
Layers	3" face brick ½" plywood R-4.9 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.2 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	4" heavyweight concrete Carpet and pad	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	5250	
Wall height, ft	10	
Internal mass, lb/ft^2	25	
Number of stories	1	
Aspect Ratio	1.0	
Ratio of exterior perimeter to floor perimeter	0.75	
Design equipment gains, W/ft^2	0.0	
Design light gains, W/ft^2	2.0	
Ave. daily min. lights/equip. gain fraction	0.2	
Ave. daily max. lights/equip. gain fraction	1.0	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	275	
CO_2 people generation, L/min-person	0.35	
Design occupancy for vent., people/1000 ft^2	30	
Design ventilation, cfm/person	20	
Average weekday peak occucpancy, ft^2/person	50	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-6 7-12 13-24	Values 0.0 0.2,0.3,0.1,0.05,0.2,0.5 0.5,0.4,0.2,0.05,0.1,0.4, 0.6,0.5,0.4,0.2,0.1,0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-6 7-12 13-24	Values 0.0 0.3,0.4,0.5,0.2,0.2,0.3 0.5,0.5,0.5,0.35,0.25, 0.5,0.8,0.8,0.7,0.4,0.2, 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.5 1.0

Table A.3. Retail Store Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.5	
Shading Coefficient	0.76	
Area ratio (window/wall)	0.15	
Exterior Wall Construction		
Layers	8" lightweight concrete R-4.8 insulation R-0.89 airspace 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") 1.25" lightweight concrete R-12 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	4" lightweight concrete Carpet and pad	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	80,000	
Wall height, ft	15	
Internal mass, lb/ft^2	25	
Number of stories	2	
Aspect Ratio	0.5	
Ratio of exterior perimeter to floor perimeter	1.0	
Design equipment gains, W/ft^2	0.4	
Design light gains, W/ft^2	1.6	
Ave. daily min. lights/equip. gain fraction	0.2	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	250	
CO_2 people generation, L/min-person	0.33	
Design occupancy for vent., people/1000 ft^2	25	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	390	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-7 8 9 10-20 21 22-24	Values 0.0 0.33 0.66 1.0 0.5 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-7 8 9 10-20 21 22-24	Values 0.0 0.33 0.66 1.0 0.5 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-12	Value 1.0

Table A.4. School Class Wing Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8” concrete block R-5.7 insulation 5/8” gypsum	
Roof Construction		
Layers	Built-up roof (3/8”) ¾” plywood R-13.3 insulation R-0.92 airspace ½” acoustic tile	
Floor		
Layers	6” heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	9600	
Internal mass, lb/ft^2	25	
Wall height, ft	10	
Number of stories	2	
Aspect Ratio	0.5	
Ratio of exterior perimeter to floor perimeter	0.875	
Design equipment gains, W/ft^2	0.3	
Design light gains, W/ft^2	2.2	
Ave. daily min. lights/equip. gain fraction	0.1	
Ave. daily max. lights/equip. gain fraction	0.95	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	200	
CO_2 people generation, L/min-person	0.3	
Design occupancy for vent., people/1000 ft^2	25	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	50	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-6 7 8-11 12-15 16 17 18 19-21 22-24	Values 0.0 0.1 0.9 0.8 0.45 0.15 0.05 0.33 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-9 10-13 14-24	Value 0.0 0.1 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.5 1.0

Table A.5. School Gymnasium Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8" concrete block R-5.7 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.3 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	7500	
Internal mass, lb/ft^2	25	
Wall height, ft	32	
Number of stories	1	
Aspect Ratio	0.86	
Ratio of exterior perimeter to floor perimeter	0.86	
Design equipment gains, W/ft^2	0.2	
Design light gains, W/ft^2	0.65	
Ave. daily min. lights/equip. gain fraction	0.0	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	550	
CO_2 people generation, L/min-person	0.55	
Design occupancy for vent., people/1000 ft^2	30	
Design ventilation, cfm/person	20	
Average weekday peak occucpancy, ft^2/person	180	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-7 8-15 16-24	Value 0.0 1.0 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-24	Value 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.1 1.0

Table A.6. School Library Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8" concrete block R-5.7 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.3 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	1500	
Internal mass, lb/ft^2	25	
Wall height, ft	10	
Number of stories	1	
Aspect Ratio	0.2	
Ratio of exterior perimeter to floor perimeter	0.75	
Design equipment gains, W/ft^2	0.4	
Design light gains, W/ft^2	1.5	
Ave. daily min. lights/equip. gain fraction	0.1	
Ave. daily max. lights/equip. gain fraction	0.95	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	250	
CO_2 people generation, L/min-person	0.33	
Design occupancy for vent., people/1000 ft^2	20	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	100	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-6 7 8-11 12-15 16 17 18 19-21 22-24	Value 0.0 0.1 0.9 0.8 0.45 0.15 0.05 0.33 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-9 10-13 14-24	Value 0.0 0.1 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.5 1.0

Table A.7. School Auditorium Characteristics

Windows		
R-value, hr-ft^2-F/Btu	1.7	
Shading Coefficient	0.73	
Area ratio (window/wall)	0.18	
Exterior Wall Construction		
Layers	8" concrete block R-5.7 insulation 5/8" gypsum	
Roof Construction		
Layers	Built-up roof (3/8") ¾" plywood R-13.3 insulation R-0.92 airspace ½" acoustic tile	
Floor		
Layers	6" heavyweight concrete	
Slab perimeter loss factor, Btu/h-ft-F	0.5	
General		
Floor area, ft^2	6000	
Internal mass, lb/ft^2	25	
Wall height, ft	32	
Number of stories	1	
Aspect Ratio	0.64	
Ratio of exterior perimeter to floor perimeter	0.85	
Design equipment gains, W/ft^2	0.2	
Design light gains, W/ft^2	0.8	
Ave. daily min. lights/equip. gain fraction	0.0	
Ave. daily max. lights/equip. gain fraction	0.9	
Sensible people gains, Btu/hr-person	250	
Latent people gains, Btu/hr-person	200	
CO_2 people generation, L/min-person	0.3	
Design occupancy for vent., people/1000 ft^2	150	
Design ventilation, cfm/person	15	
Average weekday peak occucpancy, ft^2/person	100	
Default average weekday occupancy schedule * Values given relative to average peak	Hours 1-9 10-11 12 13-14 15-24	Values 0.0 0.75 0.2 0.75 0.0
Default average weekend occupancy schedule * Values given relative to average peak	Hours 1-24	Value 0.0
Monthly occupancy scaling * relative to daily occupancy schedule	Month 1-5 6-8 9-12	Value 1.0 0.1 1.0

Table A.8. Construction Material Properties

	Conductivity (Btu/h*ft*F)	Density (lb/ft^3)	Specific Heat (Btu/lb*F)
stone	1.0416	140	0.20
light concrete	0.2083	80	0.20
heavy concrete	1.0417	140	0.20
built-up roof	0.0939	70	0.35
face brick	0.7576	130	0.22
acoustic tile	0.033	18	0.32
gypsum	0.0926	50	0.20
	Thermal Resistance ($h*ft^2*F/Btu$)		
3/4" plywood	0.93703		
1/2" plywood	0.62469		
carpet and pad	2.08		
inside air	0.67		
outside air	0.33		

APPENDIX B – BASE CASE ANNUAL SIMULATION RESULTS

The assumed base case utilizes a fixed damper position with a setup/setback control thermostat and a differential enthalpy economizer. Most commercial buildings in California employ an economizer control and therefore, it is not relevant to compare savings to systems that do not have an economizer. Annual results are presented in this section for each of the prototypical building types in all California climate zones. The results include the following quantities:

- Input air conditioner energy (AC compressor and condenser fan), kWh
- Input supply fan energy, kWh
- Peak electric demand, kW
- Input gas, Therm
- Energy consumption cost, $
- Electric demand cost, $
- Total electric cost, $
- Gas consumption cost, $
- Total system operating cost, $

Tables B.1 – B.7 show the annual results for all California climate zones assuming the base case and default building descriptions. Annual AC input power and electric demand for all building types is the lowest for CZ 1. This zone is located in the northwest coastal area of California (see Figure 5). Summer ambient temperatures are relatively moderate in this region and afford a greater opportunity for economizer controls. The highest cooling requirements are for buildings in zones 4, 10, 13 and 15. These zones are located in the south-central area of the state where it is typically very hot and dry during the summer. Not as much opportunity exists for economizer control and as a result, more mechanical cooling is required. Zones in the southwest and north/central east areas of the state require a moderate amount of mechanical cooling. Higher ambient temperatures are found here, however, not in the extreme dry bulb ranges found in the south-central zones. Input gas for furnace operation is relatively low for the entire state of California when compared to other locations across the United States. Climate zones 1, 14 and 16 typically require the most heating during winter months. These locations are further to the north and eastern areas of the state. Zones in the central and western areas of the state require less heating with generally the least amount of heating in zones 6, 7 and 15.

Table B.1. Office Base Case Annual Results

Setup/ Setback with Economizer - Base Case									
location	AC, kwh	Fan, kwh	Elec. Dmd, kW	Gas, Therm	Energy, $	Demand, $	Total Elec., $	Gas, $	Total, $
CACZ01	3497	6328	14	277	652	1583	2235	205	2439
CACZ02	15236	9570	23	322	1683	2788	4470	238	4708
CACZ03	10439	7814	18	119	1231	2231	3462	88	3550
CACZ04	17902	9395	25	169	1841	2841	4681	125	4807
CACZ05	12716	7961	19	100	1371	2306	3677	74	3751
CACZ06	17312	7931	18	17	3848	703	4552	12	4564
CACZ07	20216	8316	21	12	2783	1108	3891	9	3900
CACZ08	22873	9213	24	44	4879	897	5777	31	5808
CACZ09	24499	11365	28	28	5490	1048	6537	20	6557
CACZ10	25879	10710	27	88	5683	1036	6720	62	6782
CACZ11	22613	11314	28	354	2266	3222	5488	262	5749
CACZ12	21107	11480	29	287	2188	3175	5363	213	5576
CACZ13	30116	12567	32	224	2810	3677	6488	166	6653
CACZ14	25515	11151	28	368	5762	1039	6802	260	7062
CACZ15	49615	12966	34	30	9194	1353	10547	22	10569
CACZ16	10483	9099	22	1082	3261	792	4053	766	4819

Table B.2. Restaurant Dining Area Base Case Annual Results

Setup/ Setback with Economizer - Base Case									
location	AC, kwh	Fan, kwh	Elec. Dmd, kW	Gas, Therm	Energy, $	Demand, $	Total Elec., $	Gas, $	Total, $
CACZ01	2329	9693	15	1969	742	1523	2264	1434	3699
CACZ02	17331	19882	27	1697	2410	3204	5614	1245	6859
CACZ03	8879	15802	23	975	1567	2593	4159	717	4877
CACZ04	20949	23008	36	1031	2833	3728	6561	760	7321
CACZ05	10268	16955	24	931	1713	2643	4356	681	5037
CACZ06	16266	14309	24	383	4452	850	5302	271	5573
CACZ07	19990	16418	31	299	3539	1348	4887	224	5111
CACZ08	24522	19509	32	454	6440	1157	7596	321	7918
CACZ09	29125	23927	37	407	7747	1341	9088	288	9376
CACZ10	32998	23997	34	668	8399	1299	9698	473	10171
CACZ11	29835	23269	33	1569	3402	3595	6997	1159	8156
CACZ12	25643	22219	33	1448	3095	3523	6618	1068	7686
CACZ13	41199	26197	38	1127	4277	4257	8534	833	9367
CACZ14	34117	23868	34	1700	8554	1264	9818	1203	11022
CACZ15	72157	28763	44	294	14247	1715	15962	208	16170
CACZ16	12303	19206	25	4031	4683	937	5620	2853	8473

Table B.3. Retail Store Base Case Annual Results

Setup/ Setback with Economizer - Base Case									
location	AC, kwh	Fan, kwh	Elec. Dmd, kW	Gas, Therm	Energy, $	Demand, $	Total Elec., $	Gas, $	Total, $
CACZ01	34215	85859	167	9075	7686	16403	24089	6692	30780
CACZ02	189276	171442	279	8387	23806	32462	56269	6187	62456
CACZ03	109924	130176	238	3763	15699	26149	41849	2784	44633
CACZ04	233357	187506	356	4233	27684	36940	64623	3132	67755
CACZ05	122766	137116	241	2669	16790	26106	42897	1973	44869
CACZ06	196679	117971	235	689	46939	8413	55352	488	55840
CACZ07	233119	136999	297	436	36122	13652	49774	327	50100
CACZ08	273477	159503	308	1164	64737	11239	75976	824	76800
CACZ09	312805	203975	363	889	77227	13231	90458	630	91088
CACZ10	347519	188952	325	2169	80993	12472	93465	1536	95001
CACZ11	310053	196469	328	8661	32999	36398	69398	6407	75805
CACZ12	275714	197827	343	7303	31098	36551	67650	5403	73053
CACZ13	418430	214643	380	5636	40832	42054	82886	4170	87057
CACZ14	346838	199771	338	9049	82937	12583	95520	6406	101926
CACZ15	710167	239280	433	660	137239	16948	154187	467	154654
CACZ16	129851	152802	257	26113	44036	9281	53317	18485	71802

Table B.4. School Library Base Case Annual Results

Setup/ Setback with Economizer - Base Case									
location	AC, kwh	Fan, kwh	Elec. Dmd, kW	Gas, Therm	Energy, $	Demand, $	Total Elec., $	Gas, $	Total, $
CACZ01	612	2379	4	224	190	421	611	165	777
CACZ02	4590	4189	8	203	582	896	1477	150	1627
CACZ03	2595	3251	6	94	384	700	1084	70	1154
CACZ04	5430	4358	9	109	646	947	1593	81	1673
CACZ05	3065	3401	6	57	420	704	1124	42	1166
CACZ06	4591	3723	7	11	1238	238	1476	8	1484
CACZ07	5731	3426	7	6	894	363	1257	5	1262
CACZ08	6751	3904	8	24	1593	302	1895	17	1911
CACZ09	7538	4950	10	18	1869	362	2231	13	2244
CACZ10	8368	4589	9	43	1964	345	2309	30	2339
CACZ11	7749	4760	9	225	819	1038	1857	166	2023
CACZ12	6816	4802	9	190	766	1026	1791	140	1932
CACZ13	10407	4787	10	152	985	1160	2145	112	2258
CACZ14	8758	4772	9	205	2065	356	2421	145	2567
CACZ15	17545	5647	12	13	3361	497	3858	9	3868
CACZ16	3275	3635	7	625	1091	256	1347	442	1789

Table B.5. School Gym Base Case Annual Results

Setup/ Setback with Economizer - Base Case									
location	AC, kwh	Fan, kwh	Elec. Dmd, kW	Gas, Therm	Energy, $	Demand, $	Total Elec., $	Gas, $	Total, $
CACZ01	930	5606	21	2070	419	1647	2066	1517	3584
CACZ02	16170	11538	44	1674	1893	4861	6754	1233	7988
CACZ03	7784	8376	34	1069	1093	3564	4657	789	5446
CACZ04	18736	12324	55	1086	2109	5460	7570	803	8373
CACZ05	9720	9189	35	673	1256	3666	4922	497	5419
CACZ06	15729	10383	36	286	4034	1318	5352	203	5554
CACZ07	20531	9028	46	166	2909	1835	4744	124	4868
CACZ08	24718	10977	50	308	5560	1796	7355	218	7573
CACZ09	28146	14594	58	322	6721	2087	8808	228	9036
CACZ10	30893	13435	53	411	7057	2049	9106	291	9397
CACZ11	29218	14074	54	1723	2897	5890	8787	1275	10062
CACZ12	26250	13696	55	1575	2696	5696	8392	1165	9557
CACZ13	41168	14330	61	1280	3650	6549	10199	947	11146
CACZ14	34363	14297	55	1511	7711	2069	9780	1070	10850
CACZ15	73327	17778	71	173	13455	2723	16178	122	16300
CACZ16	11051	10144	42	3924	3585	1468	5053	2778	7831

Table B.6. School Classroom Wing Base Case Annual Results

Setup/ Setback with Economizer - Base Case									
location	AC, kwh	Fan, kwh	Elec. Dmd, kW	Gas, Therm	Energy, $	Demand, $	Total Elec., $	Gas, $	Total, $
CACZ01	5535	16231	26	382	1388	3007	4395	283	4678
CACZ02	28507	22765	43	470	3392	5131	8522	347	8870
CACZ03	18135	19092	36	136	2443	4349	6793	100	6893
CACZ04	34679	24676	52	193	3911	5760	9671	143	9814
CACZ05	22393	19751	37	88	2721	4381	7101	65	7167
CACZ06	29995	23756	40	6	7857	1453	9310	4	9314
CACZ07	38187	20338	44	3	5683	2247	7929	2	7932
CACZ08	43903	22324	50	34	9754	1754	11508	24	11532
CACZ09	46868	28725	56	13	11201	2072	13273	9	13282
CACZ10	51342	25735	50	90	11557	1931	13488	64	13551
CACZ11	44422	24917	49	579	4532	5665	10197	428	10625
CACZ12	40086	24557	50	448	4259	5618	9877	332	10209
CACZ13	59456	25356	54	348	5486	6400	11886	257	12143
CACZ14	49725	25598	50	614	11410	1917	13327	435	13762
CACZ15	99443	30298	63	22	18675	2625	21300	15	21315
CACZ16	20232	19548	39	2177	6255	1434	7689	1541	9231

Table B.7. School Auditorium Base Case Annual Results

Setup/ Setback with Economizer - Base Case									
location	AC, kwh	Fan, kwh	Elec. Dmd, kW	Gas, Therm	Energy, $	Demand, $	Total Elec., $	Gas, $	Total, $
CACZ01	219	10855	23	3739	713	1808	2521	2715	5236
CACZ02	18264	12109	69	2726	2092	7291	9382	2006	11388
CACZ03	5229	10869	45	2080	1077	4855	5932	1532	7464
CACZ04	20421	12911	93	1900	2274	8308	10582	1404	11986
CACZ05	8800	10854	54	1104	1299	5373	6672	813	7485
CACZ06	14539	10882	54	669	3909	1942	5851	473	6324
CACZ07	19222	10915	76	433	2973	2713	5686	324	6011
CACZ08	27241	11847	80	594	6183	2855	9038	420	9459
CACZ09	33231	15704	97	622	7779	3317	11096	441	11536
CACZ10	36685	15006	86	705	8366	3374	11740	499	12240
CACZ11	33972	15330	85	2699	3317	8954	12271	1994	14265
CACZ12	29786	14327	84	2607	2987	8432	11418	1926	13345
CACZ13	48705	15989	95	2097	4277	10188	14464	1551	16015
CACZ14	42047	16090	91	2392	9283	3420	12704	1693	14397
CACZ15	94136	21104	122	232	17284	4675	21959	164	22124
CACZ16	12381	11086	57	5780	3976	2163	6139	4092	10230

APPENDIX C – NEW BUILDING DESIGN APPLICATION RESULTS

Table C.1. Savings for DCV+EC in New Building Applications

	RTU Size tons	number of DCV units	First Cost $	Annual Cost Savings, $
		Office		
CACZ06	14.54	2	1800	299
CACZ15	23.91	2	1800	948
		Restaurant		
CACZ06	14.80	2	1800	446
CACZ15	29.73	2	1800	3269
		Retail Store		
CACZ06	144.82	7	6300	3775
CACZ15	294.49	14	12600	37612
		Auditorium		
CACZ06	42.54	3	2700	1921
CACZ15	78.77	4	3600	8430

Table C.2. Savings for HXHR in New Building Applications

	RTU Size tons	OA frac.	HXHR Downsize, tons	Downsize Cost Saved, $	Vent. Flow cfm	First Cost $	Annual Cost Savings, $
				Office			
CACZ06	14.01	0.188	0.53	529	921	1842	-726
CACZ15	21.07	0.125	2.85	2846	924	1848	490
				Restaurant			
CACZ06	12.98	0.692	1.82	1821	3144	6287	-1117
CACZ15	20.39	0.439	9.34	9336	3132	6264	3172
				Retail Store			
CACZ06	126.49	0.678	18.33	18333	30000	60000	-10603
CACZ15	201.78	0.424	92.71	92714	29994	59988	29054
				Auditorium			
CACZ06	24.46	0.909	18.08	18079	13497	26994	-730
CACZ15	36.98	0.909	41.79	41795	13514	27028	7177

Table C.3. Savings for HPHR in New Building Applications

	RTU Size tons	OA frac.	HXHR Downsize, tons	Downsize Cost Saved, $	Vent. Flow cfm	First Cost $	Annual Cost Savings, $
				Office			
CACZ06	14.06	0.187	0.47	474	921	4604	-860
CACZ15	21.36	0.124	2.56	2557	924	4620	6
				Restaurant			
CACZ06	11.92	0.753	2.88	2882	3144	15719	-1240
CACZ15	21.27	0.421	8.46	8458	3134	15668	1297
				Retail Store			
CACZ06	117.82	0.728	27.00	27003	30000	150000	-12700
CACZ15	212.73	0.402	81.77	81766	29998	149991	10454
				Auditorium			
CACZ06	25.10	0.886	17.44	17442	13497	67486	-1256
CACZ15	42.12	0.796	36.65	36650	13480	67402	3653

Table C.4. Cumulative Rate of Return for New Building Applications – DCV+EC

	Cumulative Years							
	0	1	2	3	4	5	6	7
				Office				
CACZ06	-100%	-83.4%	-66.8%	-50.2%	-33.6%	-16.9%	-0.3%	16.3%
CACZ15	-100%	-47.3%	5.3%	58.0%	110.7%	163.3%	216.0%	268.7%
				Restaurant				
CACZ06	-100%	-75.2%	-50.4%	-25.7%	-0.9%	23.9%	48.7%	73.4%
CACZ15	-100%	81.6%	263.2%	444.8%	626.4%	808.1%	989.7%	1171.3%
				Retail Store				
CACZ06	-100%	-40.1%	19.8%	79.8%	139.7%	199.6%	259.5%	319.4%
CACZ15	-100%	198.5%	497.0%	795.5%	1094.0%	1392.5%	1691.0%	1989.6%
				Auditorium				
CACZ06	-100%	-28.9%	42.3%	113.4%	184.6%	255.7%	326.9%	398.0%
CACZ15	-100%	134.2%	368.3%	602.5%	836.7%	1070.8%	1305.0%	1539.2%

Table C.5. Cumulative Rate of Return for New Building Applications - HXHR

	Cumulative Years							
	0	1	2	3	4	5	6	7
	Office							
CACZ06	-71.3%	-110.7%	-150.1%	-189.6%	-229.0%	-268.4%	-307.8%	-347.2%
CACZ15	54.0%	80.5%	107.0%	133.6%	160.1%	186.6%	213.1%	239.6%
	Restaurant							
CACZ06	-71%	-88.8%	-106.6%	-124.3%	-142.1%	-159.9%	-177.6%	-195.4%
CACZ15	49%	99.7%	150.3%	201.0%	251.6%	302.2%	352.9%	403.5%
	Retail Store							
CACZ06	-69%	-87.1%	-104.8%	-122.5%	-140.1%	-157.8%	-175.5%	-193.1%
CACZ15	55%	103.0%	151.4%	199.9%	248.3%	296.7%	345.2%	393.6%
	Auditorium							
CACZ06	-33%	-35.7%	-38.4%	-41.1%	-43.8%	-46.5%	-49.3%	-52.0%
CACZ15	55%	81.2%	107.7%	134.3%	160.8%	187.4%	214.0%	240.5%

Table C.6. Cumulative Rate of Return for New Building Applications – HPHR

	Cumulative Years							
	0	1	2	3	4	5	6	7
	Office							
CACZ06	-89.7%	-108.4%	-127.1%	-145.7%	-164.4%	-183.1%	-201.8%	-220.5%
CACZ15	-44.7%	-44.5%	-44.4%	-44.3%	-44.1%	-44.0%	-43.9%	-43.8%
	Restaurant							
CACZ06	-82%	-89.6%	-97.4%	-105.3%	-113.2%	-121.1%	-129.0%	-136.9%
CACZ15	-46%	-37.7%	-29.5%	-21.2%	-12.9%	-4.6%	3.6%	11.9%
	Retail Store							
CACZ06	-82%	-90.5%	-98.9%	-107.4%	-115.9%	-124.3%	-132.8%	-141.3%
CACZ15	-45%	-38.5%	-31.5%	-24.6%	-17.6%	-10.6%	-3.7%	3.3%
	Auditorium							
CACZ06	-74%	-76.0%	-77.9%	-79.7%	-81.6%	-83.5%	-85.3%	-87.2%
CACZ15	-46%	-40.2%	-34.8%	-29.4%	-23.9%	-18.5%	-13.1%	-7.7%